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SER-50959

National Aeronautics and Space Administration

OIL-AIR MIST LUBRICATION FOR HELICOPTER GEARING

By: F. McGrogan

SIKORSKY AIRCRAFT Division of UNITED TECHNOLOGIES

Prepared For

National Aeronautics and Space Administration

and U. S. Army Air Mobility Research and Development Laboratory

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16. Abstract

The operation of helicopter type spur gears with mist lubrication was explored by conducting preliminary, step-speed and endurance tests on a regenerative - closed loop test rig. For the preliminary testing at 10 krpm, the mist nozzle position, the oil/air flow ratio and tooth contact pressures (690, 1034 and 1380 Megapascals) were varied using two lubricants (a MIL-L-23699, type II ester and a formulated synthetic hydrocarbon) in order to optimize the nozzle position and oil/air flow ratio. In the step-speed tests, the speed was varied between 10 krpm and 20 krpm, with a radial nozzle placement (based on visual results of the preliminary test) and a tooth contact pressure of 1034 Megapascals. Using the same two lubricants, the heat balance data indicated a 15 to 20% increase in heat generation, but a superior tooth surface with the conventional jet spray system. The mist lubrication endurance tests were conducted at 14 krpm with a tooth pressure of 1034 Megapascals. Slight surface scuffing was present on the gear teeth at the completion of the fifty (50) hour test, the XRL-850 lubricant was graded superior. This test demonstrated the feasibility of using a once-through oil-air mist lubrication system in a two mesh gearbox.

A step-speed study, run at 10 krpm to 18 krpm and at tooth pressures of 690 and 1034 Megapascals with the two lubricants, and an endurance test run at 14 krpm and 1034 Megapascals of the emergency aspirator mist system resulted in a successful completion of a five (5) hour endurance test with the MIL-L-23699 lubricant. The success of this rather severe test demonstrated that an aspirator mist system is a good candidate for an emergency lubrication system in a helicopter transmission.

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SUMMARY

To determine the nozzle position for the Sikorsky Aircraft two mesh gearbox, a series of thirty-eight (38) individual tests were conducted at a nominal speed of 10,000 rpm. The flow of cooling air was held constant at 0.316 mol/s (15 SCFM) per mesh and was directed into the out of mesh location of the gear sets. The mist air flow was maintained at 0.063 mol/s (3 SCFM) and the amount of lubricant in the air mist was varied between 3 - 53 cc/hour. Cooling and mist air was supplied at 360K (200°F) and two lubricants were used. Based on these tests, a radial nozzle position was selected.

A comparison of mist lubrication to conventional jet spray lubrication was conducted. Two lubricants were tested at a gear contact stress of 10.34×10^8 Pa (150,000 psi) while the gear tester was operated at increasing speeds, in increments of 2,000 rpm, from 10,000 rpm to 20,000 rpm.

In the mist lubrication mode, cooling air was supplied at 366K (200°F) to the out of mesh location of the gear sets. The mist air was also supplied at 366K (200°F) to the radial position mist nozzle at a constant rate of 0.0632 mol/s (3 SCFM) per nozzle. The lubricant contained in the mist air varied between 32 - 44 cc/hour.

In the recirculating jet spray mode the flow rate was varied between 1893 - 2650 cc/hour. Visual inspection revealed the jet spray mode produced a superior surface finish on the gear teeth but a thermal energy survey showed a 15 - 20% increase in heat generated. The gear tooth condition in the mist lubrication mode system could be improved if the cooling air and lubricant/air flow ratio were increased. However, the mist lubrication system was operating at maximum capacity and higher values could not be obtained.

A heat balance for the test gearbox was established for all the aforementioned tests.

Fifty (50) hour mist lubrication tests were attempted using two lubricants. The type II ester (MIL-L-23699) lubricant test was halted after 90 minutes of operation due to a gear failure. The gears tested were old test gears and this may have influenced the test results. A successful 50 hour test using a formulated synthetic hydrocarbon (XRL-850A) lubricant was conducted at 14,000 rpm with a gear contact stress of 10.34×10^8 Pa (150,000 psi). Cooling air was supplied at 366K (200°F) at a rate of 0.316 mol/s (15 SCFM) and the mist air/lubricant ratio per nozzle was 0.063 mol/s (3 SCFM) of air at 366K (200°F) and 22.2 cc/hour of lubricant per nozzle. The latter test demonstrated the feasibility of using a once-through oil-air mist lubrication system in a two mesh gearbox as a replacement for the more vulnerable and heavier recirculating jet spray lubrication system.

An emergency lubrication system using an aspirator mist/air system was incorporated on the test gearbox and two lubricants were tested at increasing speeds, in increments of 2,000 rpm from 10,000 rpm to 18,000 rpm. The gear contact pressures tested were 6.895 Pa $\times 10^8$ (100,000 psi) and

SUMMARY (continued)

10.34×10^8 Pa (150,000 psi).

In addition, an endurance test of the emergency aspirator mist air system was conducted at 14,000 rpm using MIL-L-23699 ester lubricant. The gear contact pressure was constant at 10.34×10^8 Pa (150,000 psi) and the aspirator air was supplied at ambient temperature at a rate of 0.074 mol/s (3.5 SCFM) and an oil flow rate of 1.52 cc/minute. These results demonstrate that a simple aspirator mist system is a good candidate for an emergency lubrication system in helicopter transmission and was operable at rather severe conditions for periods up to five hours without failure. Further testing of the aspirator mist air system should be pursued.

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1.0 INTRODUCTION

A typical helicopter main transmission contains spur, helical, planetary and spiral bevel gears. These gear sets are usually supported by anti-friction bearings and lubricated by oil which is supplied under pressure from a recirculating pump. Oil is drawn from a sump and transferred through cored and/or external lines to an oil cooler. Pressurized oil is then fed to various nozzles or jets located throughout the gearbox.

Military helicopters equipped with this type of lubrication system are susceptible to sudden loss of lubricant, due to small arms fire, particularly in the area of the externally mounted oil cooler which presents a relatively large vulnerable area. A possible alternative lubrication system is a once-through oil-mist system. The principle of mist lubrication is the atomization of lubricating oil with compressed air into a fine oil mist in a mist generator and the transfer of airborne oil to the required areas. Additional air for cooling is provided separately. This program was directed to evaluate the applicability of a once-through mist system to the lubrication of helicopter spur gears. A complete evaluation of bearing lubrication, wear rates, debris removal, fatigue corrosion and seal problems would require long term tests and was not considered in this program.

The mist lubrication concept is an accepted technique for low speed applications particularly in the machine tool trade. It offers the potential advantages of increased gearbox efficiency. Since, in a mist lubrication system, a very fine fog is used for actual gear lubrication, and air is used as the primary system coolant, the losses must be balanced against the necessary power required to provide an adequate flow of cooling air. It has been established that mist lubrication of a high performance aircraft gear rig (i.e., Ryder Gear Tester) is technically feasible (see Appendix 2). However, practical applicability to aircraft transmission will depend both on lubrication performance and on power requirements for cooling the gears.

This report covers the work performed under NASA Contract NAS3-18538. This work is closely related to other recent NASA studies on Mist Lubricant Application Systems, (1, 2, 3, 4, and 5) but differs since the tests are conducted on actual gears in a dynamic system, under loads and speeds which simulate helicopter transmission operating conditions. Preliminary system studies were conducted for NASA Contract NAS3-16825 (see Appendix 2) to determine optimized mist lubrication parameters for once-through lubrication on a simulated helicopter spur gear mesh.

The object of this program was to apply the same mist lubrication parameters to the Sikorsky Aircraft test gearbox and expand the tests previously conducted for NASA Contract NAS3-16825 (see Appendix 2).

The optimized mist lubrication system selected from the initial tests was then applied to the test gearbox at various speeds using two lubricants supplied by NASA. Tests were repeated using a conventional recirculating spray lubrication system.

The heat generated in the test box was obtained and a heat balance was established for each test run. An analysis of the significant performance parameters was conducted.

Based on test experience gained and the subsequent data generated, a severe test condition was selected and 50 hour endurance tests of the mist lubrication system were attempted using the two test lubricants.

Study of an emergency lubrication system, using an aspirating device was also conducted. Tests of the emergency lubricating system at various speeds and two lubricants were run. A five (5) hour endurance test of this system was conducted at a severe test condition and with one lubricant.

2.0 TEST EQUIPMENT

2.1 Sikorsky Test Rig

A Sikorsky designed test facility, incorporating a closed loop regenerative test fixture, Figure 1, was used to evaluate the lubrication systems. In this test facility, two gearboxes, each containing two sets of test gears, are connected by shafts mounted on flexible couplings which reduce interactions between the two gearboxes. This arrangement permits up to four sets of gears to be tested simultaneously (see Figure 2).

The test gears are piloted on the outboard end of the gearbox shafts. This configuration permits ready access for gear inspection and replacement. Torque is transmitted to the gears by four close tolerance bolts which also retain the gears on the shafts. A 59.7 Kilowatt (80 hp) varidrive electric motor supplies the necessary drive power to overcome the friction of the system. A vee belt drive transmits the power to the closed loop system via a spur gear set with a 3.3:1 ratio connected to the slave test box.

Torque is applied to the system by the relative angular displacement of vernier plates on one of the connecting shafts. Strain gages bonded in a torque sensing orientation to the connecting drive shafts and wired to a SR-4 strain indicator are used to measure system torque while the load is being applied. System windup provides adequate sensitivity to obtain the desired torque levels.

Each gearbox is equipped with an independent lubrication system which operates at flow rates of up to $0.019 m^3/min$ (5 gpm). The oil reservoir in each system has a capacity of $0.057 m^3$ (15 gallons). A 40-micron filter on each supply line maintains oil cleanliness and prevents oil jet blockage.

A failure detection system which automatically shuts down the test facility when a failure occurs is installed as part of the test facility control circuitry. A low oil pressure switch protects the facility from failures due to malfunctioning oil pumps, ruptured oil lines, or low oil level in either sump. Excessive oil temperature also activates the shutdown system. Magnetic type chip detectors are incorporated to stop the test if metallic particles enter the lubrication system.

A missing tooth indicator was used when operating the gear tester at 10,000 rpm. This device compares an input signal from a magnetic tooth contactor (on a cycle per cycle basis) to an internal signal generated by an oscillator which is phase locked to the contactor signal. If a tooth is missing, the comparison on that cycle triggers a flip-flop which trips the motor relay to shut off the machine. The time from detection to relay shutoff is approximately equal to the relay closing time. There is one circuit for each

FIGURE 1 SIKORSKY FOUR INCH GEAR TESTER

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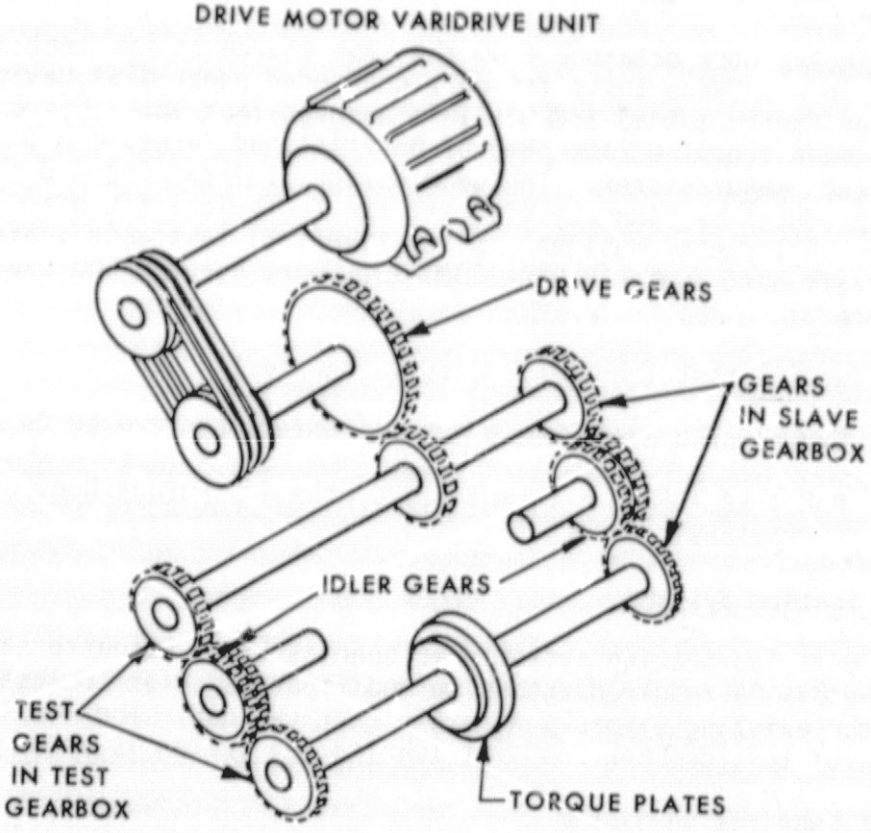

FIGURE 2 CLOSED LOOP REGENERATIVE GEAR TEST RIG

of the eight gear positions. When a bending failure occurs, which results in the loss of a gear tooth, lights on the instrument panel will indicate not only which gearbox is affected but in which of the four possible gear positions the failed gear can be found. If the test machine shuts down because of low oil pressure, low oil level, chip detection, or a recorder malfunction (high temperature), this information will also be indicated by an appropriate light on the instrument panel.

2.2 Gears and Lubricants

Test gears were obtained from the same source as the test gears used in the Exxon tests for work conducted under NASA Lewis Research Center Contract NAS3-16825. This ensured that both sets of gears were made from the same heat of material and forged and machined by the same manufacturer. The gear design, tolerancing and tip relief was produced by Sikorsky Aircraft and is typical for gears used in modern helicopter designs. Sixty-five (65) test gears, Sikorsky Aircraft part number 61050-35080-102, were ordered and used (see Figure 3).

The two (2) test lubricants used in the testing were:

(a) Exxon Turbo-oil 2380, a type II ester, conforming to MIL-L-23699

(b) Mobil XRL-850A, a formulated synthetic hydrocarbon, used in a previous contract (4).

2.3 Lubrication Systems

The mist lubricating system developed at Exxon Research Laboratories (Appendix 2) required extensive modification prior to installation on the existing Sikorsky Aircraft test gearbox. The final configuration of test gearbox and mist lubricating system is shown in Figure 4.

2.3.1 Mist Lubrication System

The Sikorsky test gearbox has a two mesh system and consequently required higher oil feed rates than were used by Exxon for their one mesh test box. Oil consumption was not known until completion of each test when the pretest and post-test oil quantities were compared. A schematic of the final plumbing arrangement is shown in Figure 5.

The oil mist generator used was a Norgren type part number 10-015-002.

The oil mist nozzles used were supplied by Exxon Research Laboratories and are depicted in Figure 6. The mist air was supplied to the Norgren generator through a rotometer and a Chromalox MTO-220A heater. Temperature control was maintained by a thermostatically activated Variac at 366K (200°F).

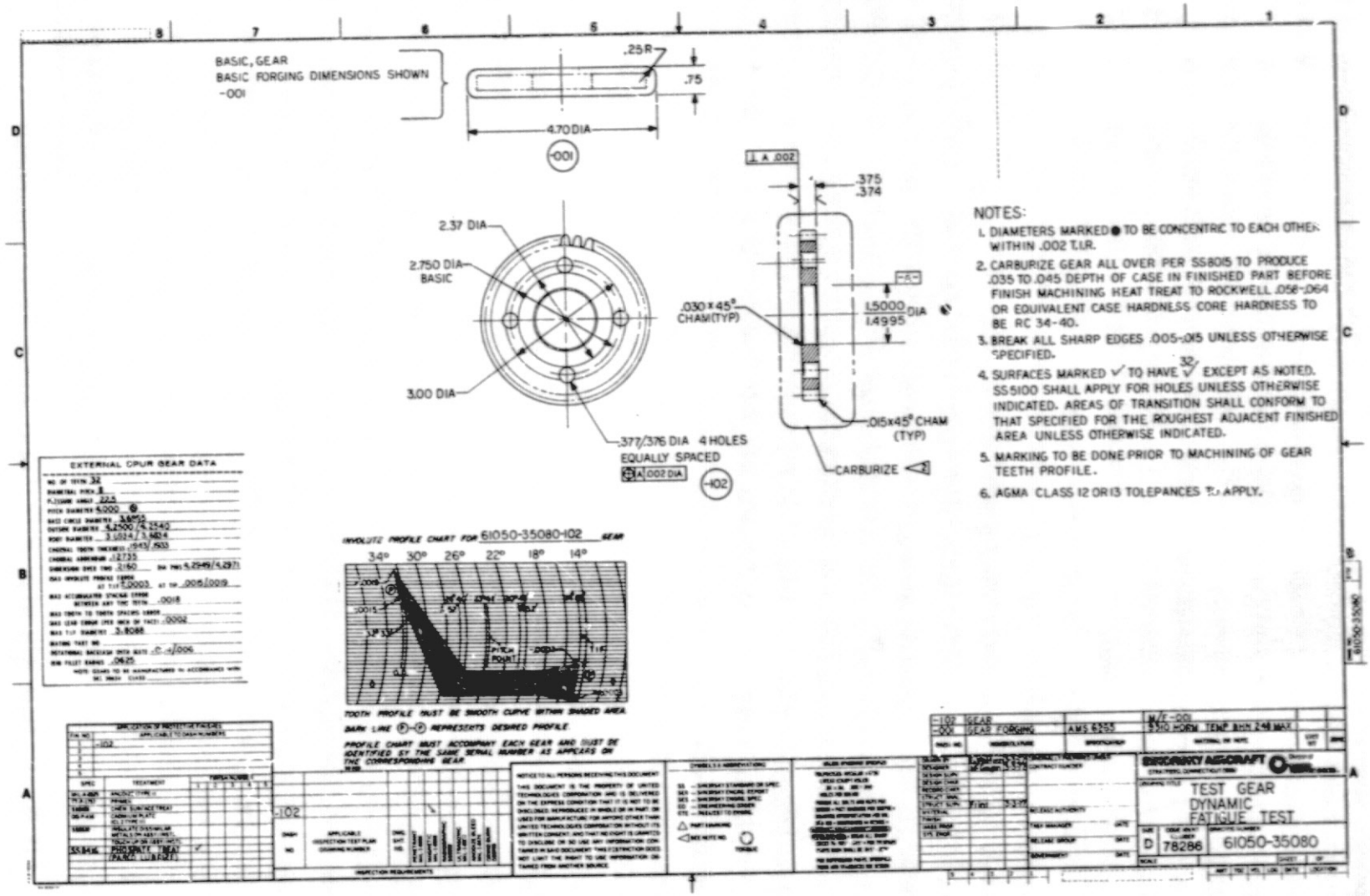

FIGURE 3 TEST GEAR - FOUR INCH DYNAMIC GEAR TESTER

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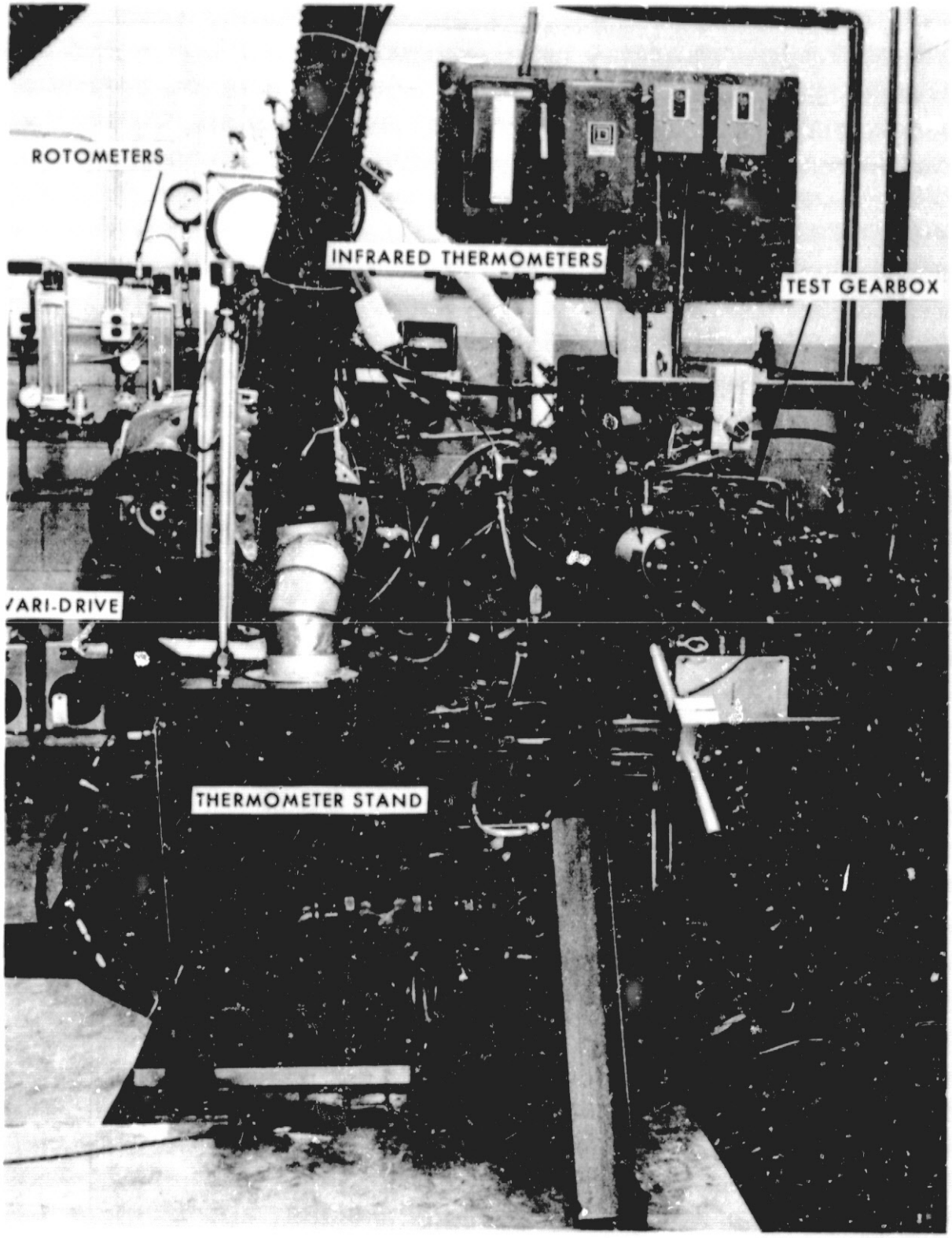
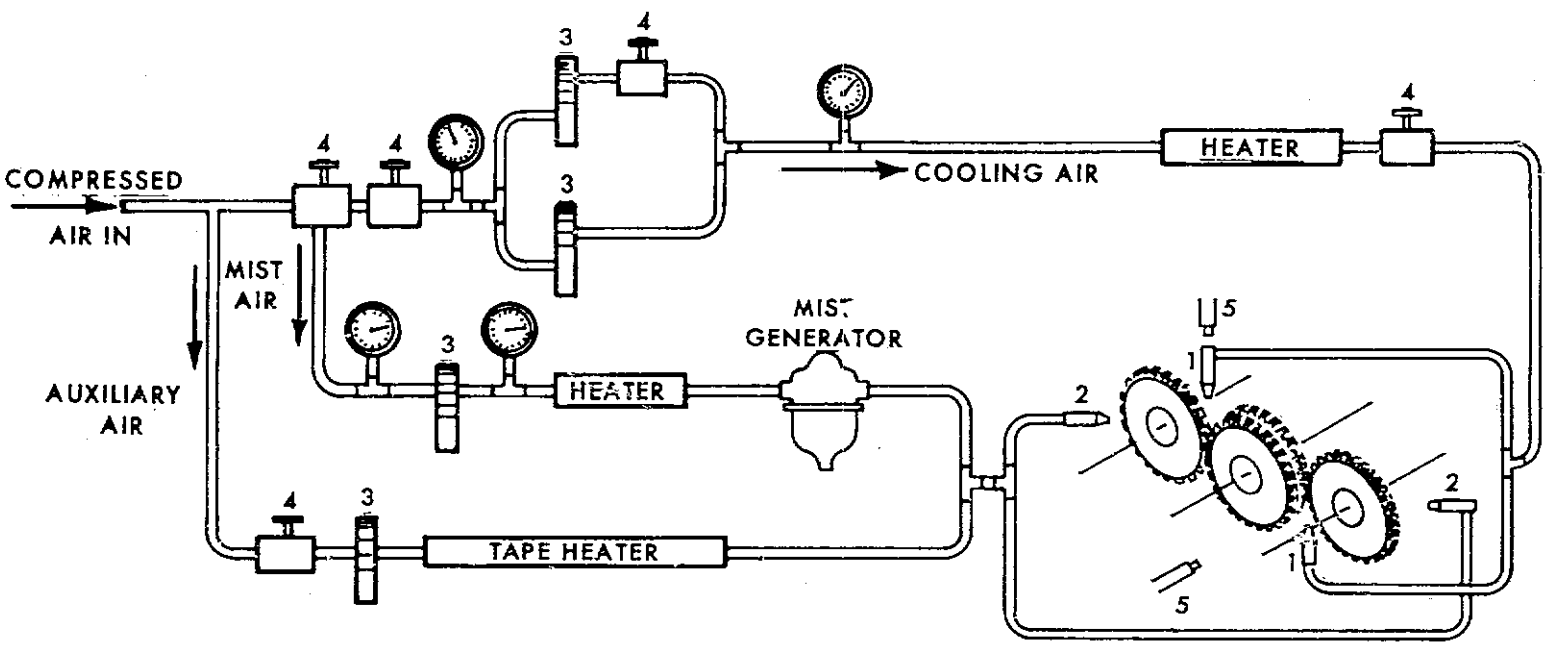
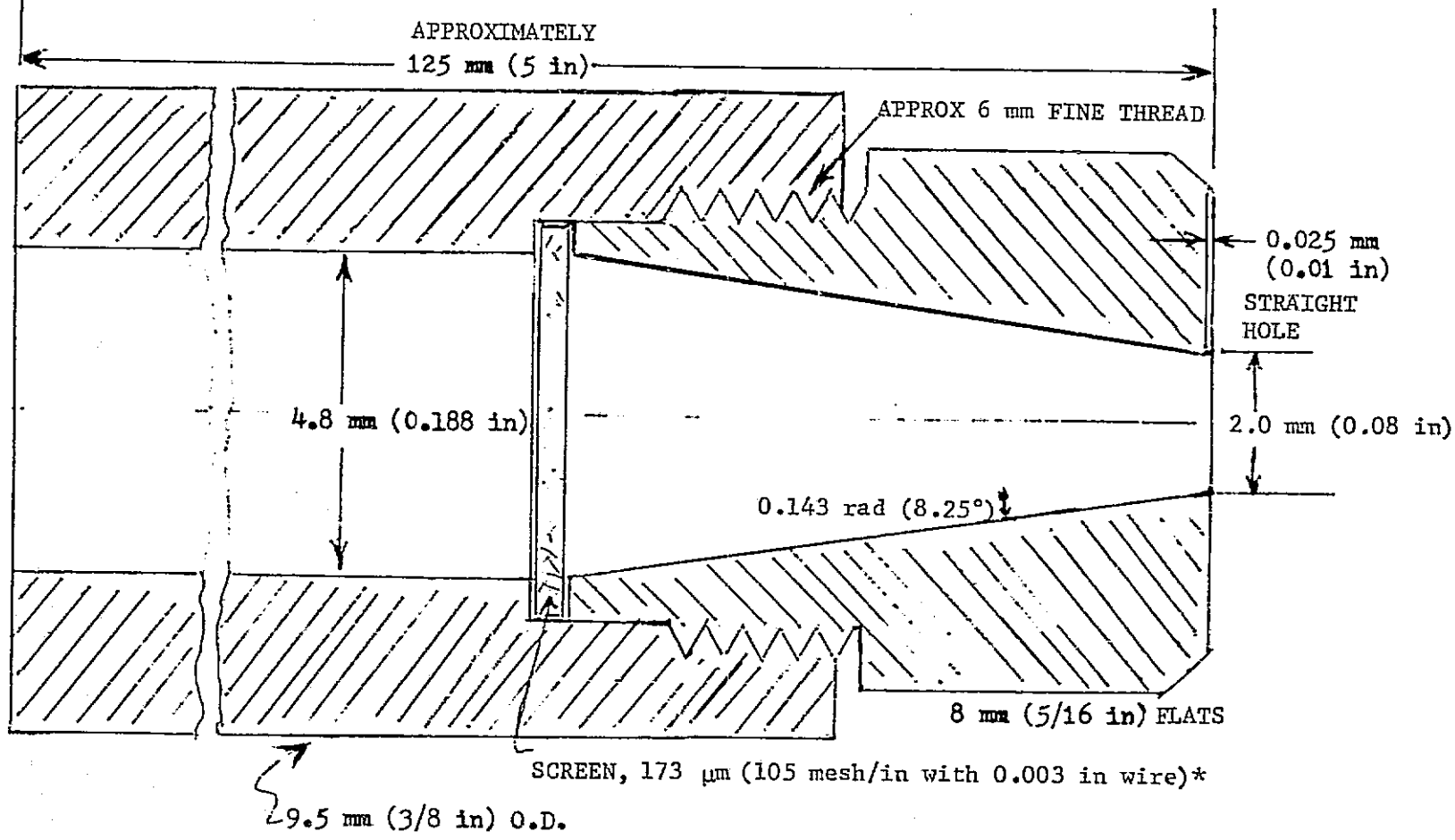


FIGURE 4 TEST RIG WITH MIST LUBRICATION SYSTEM ATTACHED



1 – COOLING AIR NOZZLE
2 – MIST CLASSIFIER NOZZLE
3 – ROTOMETER
4 – AIR VALVES
5 – INFRARED THERMOMETERS

FIGURE 5 MIST LUBRICATION AIR DELIVERY SYSTEM - SCHEMATIC



*Cambridge Wire Cloth Co., 93316-A1050-0030A-24000

FIGURE 6 NASA MIST NOZZLE

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Auxiliary air and the air/oil mist were combined prior to the nozzles to achieve 0.053 to 0.063 mol/s (2.5 to 3 SCFM) total flow at each nozzle. Auxiliary air flow was measured using a rotometer. Tape heaters, wrapped around the supply line, permitted an increase in air temperature to 366K (200°F).

Separate cooling air, which was fed through a 2,000 watt heater to maintain a 366K (200°F) temperature, was directed to impinge each gear at the "out of mesh" area. Total cooling air flow rate was 0.316 mol/s (15 SCFM) per mesh.

Temperatures were measured using iron-constantan thermocouples and were continually recorded on a strip recorder. The thermocouples were located on the air-in, air-out, oil-in, oil-out lines, adjacent to bearings, and at other strategic locations. Infrared thermometers, supplied by Exxon, were used to measure the apparent tooth and bulk temperatures of the test gears. The Exxon-supplied tripod stands for the mounting of the infrared thermometers could not be adapted to the Sikorsky test box. A heavy, wide based angle iron frame was constructed to custom mount both units in a rigid assembly. Sight tubes using a calcium fluoride window were positioned in the test gearbox housing and cover to allow the infrared thermometers to focus on the selected test gear.

A Cox flow turbine installed in each oil input line was used to measure oil flow rates. Pump output and last jet oil pressures were measured to monitor pump operation and gearbox pressure drops.

2.3.2 Aspirator Mist Lubrication System

The emergency aspirator mist lubrication system installed on the test gearbox used a graduated container placed beneath the test gearbox to simulate a limited capacity oil reservoir. Two aspirator nozzles were bonded into the test gearbox and shop air was connected to the nozzles and oil reservoir by rubber and copper tubing. The aspirator nozzle was obtained from a commercial supplier of medical spray devices and the approximate dimensions of the nozzle are depicted in Figure 19. A schematic of this system is shown also in Figure 19.

2.4 CALIBRATION

2.4.1 Test Gearbox

Thermal calibration of the test gearbox was performed in order to determine a practical heat leakage coefficient, K, for use in subsequent heat balance computations. Gears were removed from the test gearbox and electrical heater rings inserted. The ventilating system, normally in use during testing, was operated. The electrical input to the heaters recorded along with gearbox case temperatures, Tc, and ambient temperature, Ta. When the temperatures stabilized, the data was recorded and the heat leakage coefficient for the test

box was determined. The recorded run consumed 400 watts with Tc = 340K (152.5°F) and Ta = 293K (68°F). From this an equation for evaluating the heat loss due to conduction and radiation was established for the test gearbox.

$$Q_5 = \overline{K} (Tc - Ta) \text{ watts/K} \quad \text{Where } \overline{K} = 8.53 \text{ watts/K}$$

or

$$Q_5 = \overline{K} (Tc - Ta) \text{ btu/min F} \quad \text{Where } \overline{K} = 0.27 \text{ btu/min F}$$

2.4.2 Test Installation

Thermal calibration of the complete test installation was performed to evaluate the efficiency of the electric motor and varidrive.

The slave and test gearboxes were operated at various loads and speeds using a conventional recirculating jet spray lubrication system and MIL-L-23699 lubricant. Each test was conducted until temperatures stabilized (1 hour maximum). A heat balance was conducted for each condition. The operating conditions and results are tabulated in Table 1A & 1B with a graphical representation of the data shown in Figure 7. A schematic representation of the heat balance, used to determine the test rig calibration is shown in Figure 8 and a sample calculation of the heat transfer data is shown in Appendix 1.

2.4.3 Infrared Thermometers

Attempts to calibrate the infrared thermometers proved to be both time consuming and unsuccessful. The calibration technique used by Exxon Research Laboratories was initially employed. A gear, with a thermocouple attached, was heated on a hot plate in a nitrogen environment. Gear temperature was measured by the thermocouple while the infrared thermometer output was recorded using the calcium fluoride windowed sight tube used in the tests. Readings were taken over a range of emissivity settings. However, the infrared thermometer and actual temperature data could not be correlated within a reasonable degree of confidence.

The infrared thermometers were examined by the manufacturer, found to be defective and reworked. The calibration method was modified to account for the changing emissivity of the rotating gear due to a combination of black gear tooth top land and root surface, with bright steel on the tooth flank. The gear was heated in an oven and temperature readings from thermocouples on the oven and the infrared thermometer readout temperatures were recorded. No satisfactory correlation between infrared thermometer output and oven temperature was obtained with the new calibration procedure and it was decided to utilize the instruments in a purely qualitative function as an indication of possible impending gear failure.

TABLE 1A TEST GEARBOX CALIBRATION TEST RESULTS (SI UNITS)

Test Run	Speed krpm	Temperature Gearbox Tc (K)	PLV M/S	Contact Stress (Pa) x 10^8	Qin Input (Watts)	Q_1 Service Oil Test Box (Watts)	Q_4 Leakage Varidrive (Watts)	Q_5 Leakage Test Box (Watts)	Test Time (Minutes)
C_1	10	311	53.20	0	9,277	1,649	7,502	128	60
C_2	10	131	53.20	6.895	9,397	1,645	7,624	128	60
C_3	10	319	53.20	10.34	10,596	2,165	8,260	171	30
C_4	10	325	53.20	13.79	11,796	2,868	8,724	204	30
C_5	12	319	63.84	0	10,996	2,211	8,638	147	60
C_{5A}	12	327	63.84	10.34	12,995	1,848	10,929	218	30
C_6	14	323	74.48	0	13,795	2,569	11,065	161	60
C_{6A}	14	336	74.48	10.34	15,394	2,274	12,878	242	25
C_7	16	338	85.12	0	16,994	2,478	14,288	228	45
C_{7A}	16	345	85.12	10.34	18,393	2,934	15,183	266	30
C_8	18	347	95.76	0	20,794	2,975	17,556	261	30
C_{8A}	18	352	95.76	10.34	22,389	3,274	18,806	308	25

PLV = Pitch Line Velocity

TABLE 1B TEST GEARBOX CALIBRATION TEST RESULTS (CUSTOMARY UNITS)

Test Run	Speed krpm	Temperature Gearbox Tc (F)	PLV (ft/min)	Contact Stress (psi)	Q_{in} Input (BTU/Min)	Q_1 Service Oil Test Box (BTU/Min)	Q_4 Leakage Varidrive (BTU/Min)	Q_5 Leakage Test Box (BTU/Min)	Test Time (Minutes)
C_1	10	101	10,472	0	527.90	93.84	426.77	7.29	60
C_2	10	104	10,472	100,000	534.75	83.60	433.84	7.29	60
C_3	10	114	10,472	150,000	602.99	123.20	470.07	9.72	30
C_4	10	125	10,472	200,000	671.26	163.20	496.45	11.61	30
C_5	12	114	12,566	0	625.75	125.80	491.58	8.37	60
C_{5A}	12	130	12,566	150,000	739.50	105.14	621.96	12.42	30
C_6	14	122	14,661	0	785.03	146.20	629.66	9.18	60
C_{6A}	14	145	14,661	150,000	876.04	129.43	732.84	13.77	25
C_7	16	149	16,755	0	967.06	141.02	813.08	12.96	45
C_{7A}	16	162	16,755	150,000	1,046.70	166.98	864.00	15.12	30
C_8	18	166	18,850	0	1,183.23	169.30	999.08	14.85	30
C_{8A}	18	175	18,850	150,000	1,274.10	186.33	1,070.20	17.55	25

PLV = PITCH LINE VELOCITY

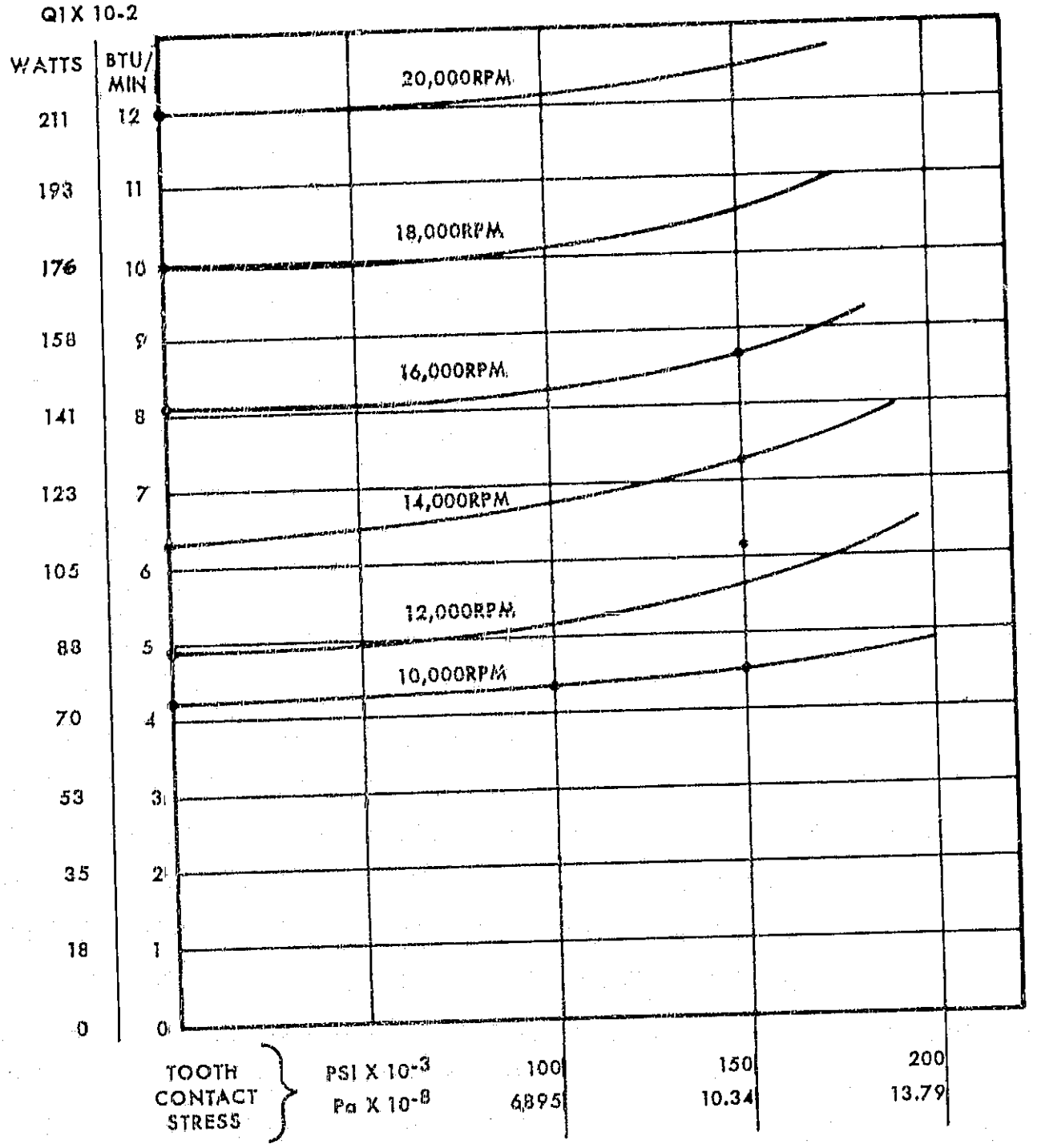

FIGURE 7 CALIBRATION CURVES FOR POWER LOSSES IN VARIDRIVE AND SLAVE GEARBOX

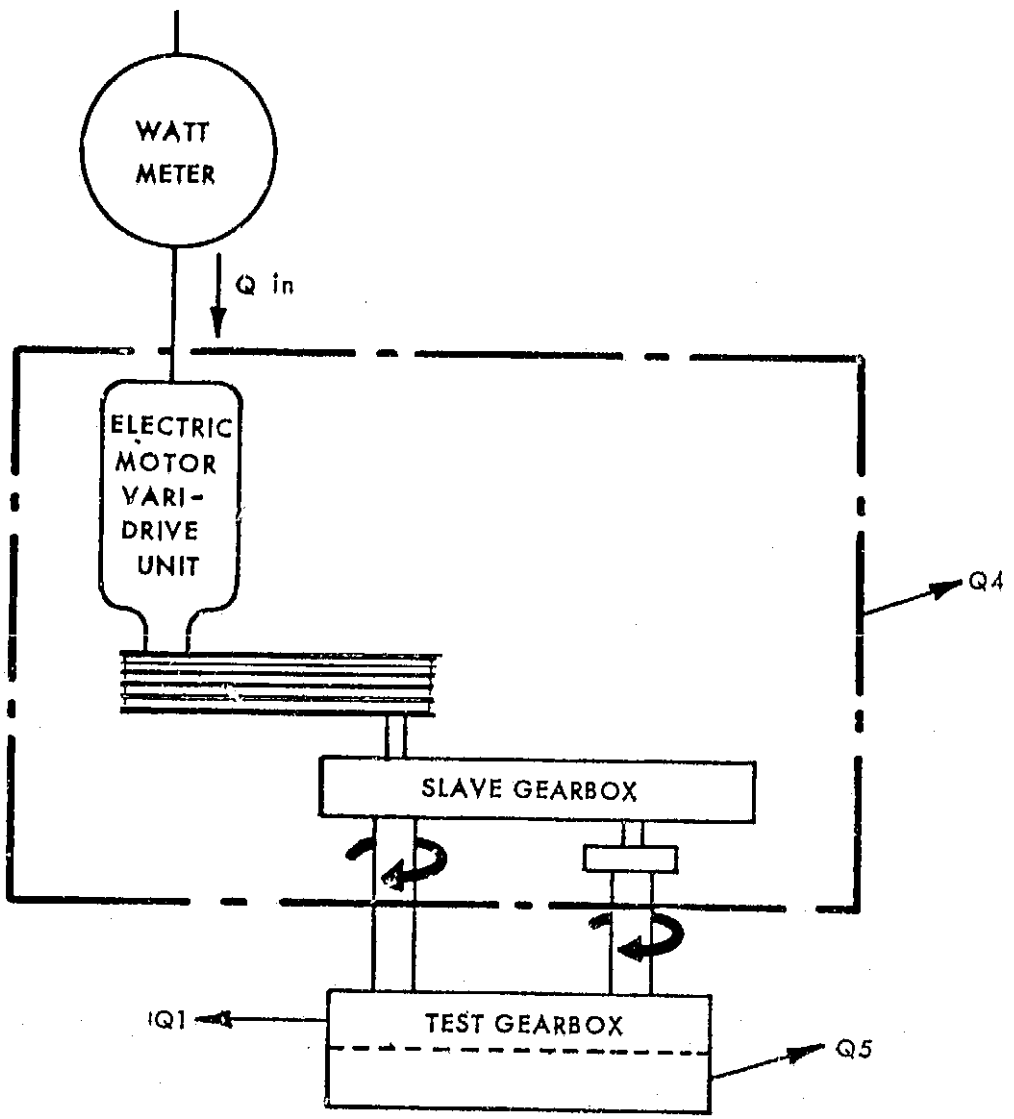

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FIGURE 8 HEAT BALANCE SCHEMATIC CALIBRATION TEST RUN

3.0 TEST PROCEDURE AND RESULTS

3.1 Preliminary Mist Lubrication Tests

Preliminary tests were conducted to determine the optimum mist nozzle position and the air/oil ratio for the Exxon supplied mist lubrication system operating on the Sikorsky two (2) mesh gearboxes.

3.1.1 Test Procedure

A conventional oil jet lubrication system was used in the slave gearbox and for the test gearbox service bearings. The test gearbox was fitted with the mist lubrication system (see Figure 4). One mist nozzle position was permanently set in the radial position for one set of gears, while the second gear mesh was lubricated by a moveable nozzle so that the following positions were obtainable (see Figure 9):

(a) Radial position 165° before mesh point

(b) Axial position 90° before mesh point with nozzle canted 15° in the direction of gear rotation

(c) Tangential position 68° before mesh point

A series of 38 individual tests were conducted at 10,000 rpm with cooling air flow constant at 0.316 mol/s (15 SCFM) per mesh and mist air constant at 0.063 mol/s per mesh. Nozzle position, tooth loading, lubricant/air mass flow ratios were varied using two lubricants. Heat flows were calculated for each test and are shown in Table 2A (SI units), Table 2B (Customary units) for MIL-L-23699 and Table 3A (SI units), Table 3B (Customary units) for Mobil XRL-850A lubricant. A sample calculation is included in Appendix 1 and a schematic of the heat balance is depicted in Figure 10. Test gears were visually inspected at the termination of each test.

3.1.2 Results and Discussion

Initially MIL-L-23699 oil from the same batch number as used for the Exxon tests was to be used since sufficient quantities of the original oil were not available MIL-L-23699 oil from a similar batch was used.

A calcium fluoride window on the infrared thermometer sight tube was found to be defective causing cooling air leaks. A new calcium fluoride window was fitted and attempts to recalibrate the corresponding infrared thermometer highlighted the inaccuracies of the temperature readout. The infrared thermometer temperature measurements were not used as "actual" temperatures but as reference temperatures in ascertaining impending gear failure.

Testing at 13.79×10^8 Pa (200,000 psi) tooth pressure caused extensive damage to the test rig when gear tooth failure occurred.

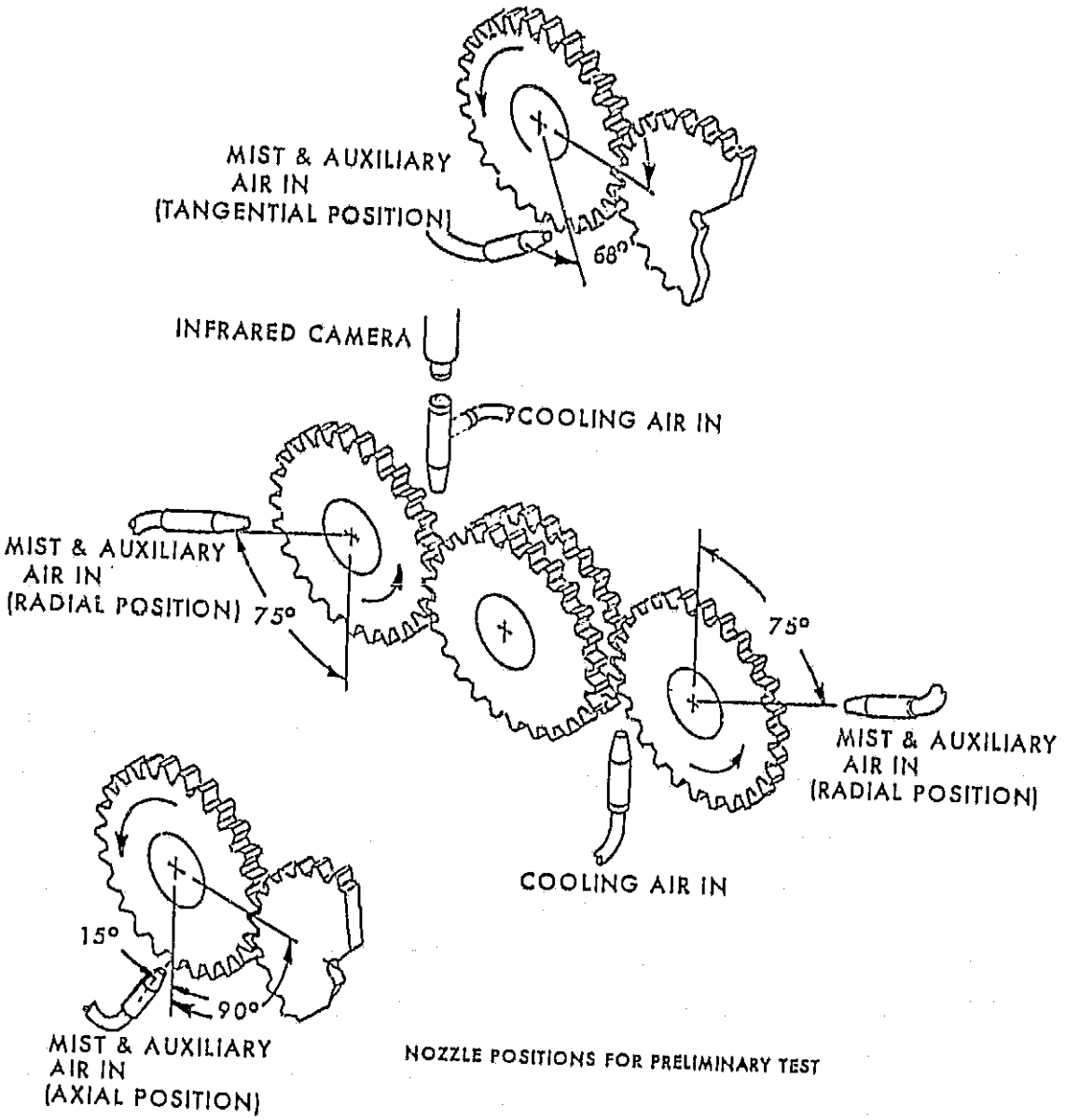

FIGURE 9 NOZZLE POSITIONS FOR PRELIMINARY TESTS

TABLE 2A PRELIMINARY TEST RESULTS - MIST LUBRICATION (SI UNITS)

Lubricant MIL-L-23699 Speed 10,000 rpm 0.316 mol/s Cooling Air Per Mesh
0.0632 mol/s Total Mist Air Per Mesh

Run No.	Gear Contact Stress (Pax10-8)	Nozzle Position **	Temperature Gearbox Tc (K)	Q_{in} Input (Watts)	Q_1 Service Test Box (Watts)	Q_2 Air (Watts)	Q_4 Leakage Varidrive (Watts)	Q_5 Leakage Test Box (Watts)	Q_{out} Output (Watts)	Gross Balance %	W_4 Oil cc/hr	Test Time Minute
1	6.895	T	273	7,397	2,307	- 620	7,556	256	9,499	128	19.0	60
2	6.895	T	420	7,397	2,417	- 821	7,556	270	9,423	127	26.0	60
3	6.895	T	352	7,397	2,802	- 718	7,556	318	9,958	135	3.0	60
4	6.895	A	352	7,597	934	- 594	7,556	370	8,267	109	6.0	60
5	6.895	A	346	7,797	1,373	-1,341	7,556	332	7,920	102	10.0	26
6	6.895	A	347	7,797	1,373	-1,294	7,556	346	7,981	102	11.5	25
7*	6.895	A	341	8,197	989	-1,208	7,556	370	7,707	94	---	25
7R	6.895	R	347	7,797	1,154	- 639	7,556	389	8,260	106	3.5	30
8	6.895	R	345	7,797	1,209	- 889	7,556	365	8,241	106	9.0	25
9	6.895	R	350	7,597	1,264	- 830	7,556	399	8,388	110	13.0	30
10*	10.34	T	364	13,594	1,099	- 137	8,171	584	9,716	72	4.0	13
11*	10.34	T	---	---	---	---	8,171	---	---	---	15.0	2
12*	10.34	T	360	16,993	1,154	- 369	8,171	536	9,493	56	15.0	10
13*	10.34	A	364	16,793	165	21	8,171	555	8,912	52	20.0	11
14*	10.34	A	366	10,996	110	22	8,171	560	8,863	81	31.0	30
15*	10.34	A	366	10,996	55	17	8,171	565	8,808	80	37.5	30
16	10.34	R	344	18,393	1,428	32	8,171	394	10,026	55	12.0	30
17	10.34	R	349	9,995	110	- 397	8,171	446	8,330	83	35.0	30
27*	13.79	R	326	19,992	-	- 336	8,786	223	8,673	43	50.0	6
27R+*	13.79	R	342	19,992	-55	- 168	8,786	342	8,905	45	21.0	3

+ Increased Cooling Air Flow 0.421 mol/s
** T = Tangential A = Axial R = Radial
* Gear Scored or Failure

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TABLE 2B PRELIMINARY RESULTS MIST LUBRICATION (CUSTOMARY UNITS)

Lubricant - MIL-L-23699 Speed - 10,000 rpm 15 scfm Cooling Air/Mesh
3 scfm Total Mist Air/Mesh

Run No.	Gear Contact Stress (kpsi)	Nozzle Position **	Temp. Gearbox Tc (F)	Q_{in} Input (Btu/Min)	Q_1 Service Oil Testbox (Btu/Min)	Q_2 Air (Btu/Min)	Q_4 Leakage Vari-drive (Btu/Min)	Q_5 Leakage Testbox (Btu/Min)	Q_{out} Output (Btu/Min)	Gross Balance %	W_4 Oil (oz/hr)	Test (Mins)
1	100	T	161	420.94	131.30	-35.29	430	14.58	540.59	128	0.643	60
2	100	T	167	420.94	137.55	-46.71	430	15.39	536.23	127	0.879	60
3	100	T	174	420.94	159.43	-40.87	430	18.09	566.65	135	0.101	60
4	100	A	175	432.32	53.14	-33.78	430	21.06	470.42	109	0.203	60
5	100	A	163	443.70	78.15	-76.33	430	18.90	450.72	102	0.338	26
6	100	A	166	443.70	78.15	-73.66	430	19.71	454.20	102	0.389	25
7*	100	A	155	466.45	56.27	-68.73	430	21.06	438.60	94	0.304	25
7R	100	R	166	443.70	65.65	-47.73	430	22.14	470.06	106	0.118	30
8	100	R	162	443.70	68.77	-50.57	430	20.79	468.99	106	0.304	25
9	100	R	171	432.32	71.90	-47.24	430	22.68	477.34	110	0.440	30
10*	150	T	195	773.62	62.52	- 7.82	465	33.21	552.91	72	0.135	13
11*	150	T	---	---	---	---	465	---	---	---	0.507	2
12*	150	T	188	967.03	65.65	-20.97	465	30.51	540.19	56	0.507	10
13*	150	A	195	955.65	9.38	1.18	465	31.59	507.15	52	0.676	11
14*	150	A	199	625.72	6.25	1.27	465	31.86	504.38	81	1.048	30
15*	150	A	199	625.72	3.13	0.99	465	32.13	501.25	80	1.268	30
16	150	R	160	1046.67	81.28	1.83	465	22.41	570.52	55	0.406	30
17	150	R	169	568.84	6.25	-22.59	465	25.38	474.04	83	1.184	30
27*	200	R	127	1137.68	---	-19.11	500	12.69	493.58	43	1.691	6
27R+*	200	R	157	1137.68	-3.13	- 9.57	500	19.44	506.74	45	0.710	3

\+ Increased cooling air flow 20 scfm/mesh
* Gear scored or failure
** T = Tangential A = Axial R = Radial

TABLE 3A PRELIMINARY TEST RESULTS MIST LUBRICATION (SI UNITS)

Lubricant XRL 850A Speed 10,000 rpm 0.316 mol/s Cooling Air per Mesh
0.0632 mol/s Total Mist Air per Mesh

Run No.	Gear Contact Stress (Pa x10^8)	Nozzle Position **	Temp. Gearbox Tc (K)	Q_{in} Input (Watts)	Q_1 Service Oil Testbox (Watts)	Q_2 Air (Watts)	Q_4 Leakage Varidrive (Watts)	Q_5 Leakage Testbox (Watts)	Q_{out} Output (Watts)	Gross Balance %	W_4 Oil cc/hr	Test Time Min.
28	6.895	T	341	8,796	49	-724	7,556	304	7,185	82	9	30
29	6.895	T	339	8,796	103	-813	7,556	318	7,164	81	15	30
30	6.895	T	339	8,796	51	-893	7,556	299	7,013	80	29	30
31	6.895	A	337	8,796	52	-892	7,556	308	7,025	80	7	30
32	6.895	A	340	8,796	51	-911	7,556	299	6,995	80	11	30
33	6.895	A	339	8,597	153	-1,745	7,556	285	6,248	73	32	30
34	6.895	R	341	8,796	0	-691	7,556	313	7,178	82	3	30
35	6.895	R	338	8,796	48	-687	10,719	318	7,235	82	11	30
36+	6.895	R	347	11,394	103	-555	7,908	375	10,265	90	53	30
37	10.34	T	350	9,596	106	-782	8,171	394	7,889	82	15	30
38	10.34	T	347	9,996	110	-860	8,171	399	7,820	78	26	30
39	10.34	T	350	9,596	103	-753	8,171	375	7,896	82	39	30
41*	10.34	A	334	19,192	52	-1,110	8,171	280	7,392	39	26	30
42*	10.34	A	352	9,796	53	-765	8,171	403	7,863	80	45	30
43*	10.34	R	349	9,596	103	-767	8,171	380	7,880	82	14	30
44	10.34	R	349	9,445	52	-790	8,171	399	7,832	83	19	30
45	10.34	R	349	9,796	-52	-616	8,171	365	7,869	80	38	30
54*	13.79	R	342	19,592	258	-934	8,786	351	8,461	43	30	5

\+ Run conducted at 12,300 rpm
* Gear scored or failed
** T = Tangential R = Radial A = Axial

TABLE 3B PRELIMINARY TEST RESULTS MIST LUBRICATION (CUSTOMARY UNITS)

Lubricant - XRL-850A Speed 10,000 rpm 15 scfm Cooling Air per Mesh
3 scfm Total Mist Air per Mesh

Run No.	Gear Contact Stress (kpsi)	Nozzle Position **	Temp. Gearbox Tc (F)	Q_{in} Input (Btu/Min)	Q_1 Service Oil Testbox (Btu/Min)	Q_2 Air (Btu/Min)	Q_4 Leakage Vari-drive (Btu/Min)	Q_5 Leakage Testbox (Btu/Min)	Q_{out} Output (Btu/Min)	Gross Balance %	W_4 Oil (oz/hr)	Test Time (Mins)
28	100	T	155	500.58	2.79	-41.20	430	17.28	408.87	82	0.304	30
29	100	T	150	500.58	5.86	-46.28	430	18.09	407.67	81	0.507	30
30	100	T	150	500.58	2.90	-50.84	430	17.01	399.07	80	0.981	30
31	100	A	148	500.58	2.93	-50.73	430	17.55	399.75	80	0.237	30
32	100	A	152	500.58	2.90	-51.83	430	17.01	398.08	80	0.372	30
33	100	A	151	489.20	8.69	-99.32	430	16.20	355.57	73	1.082	30
34	100	R	155	500.58	0	-39.33	430	17.82	408.49	82	0.101	30
35	100	R	149	500.58	2.75	-39.11	610	18.09	411.73	82	0.372	30
36+	100	R	165	648.40	5.86	-31.56	450	21.33	584.13	90	1.792	30
37	150	T	170	546.09	6.01	-44.50	465	22.41	448.92	82	0.507	30
38	150	T	166	568.84	6.23	-48.91	465	22.68	445.00	78	0.879	30
39	150	T	171	546.09	5.86	-42.84	465	21.33	449.35	82	1.319	30
41*	150	A	142	1,092.17	2.93	-63.19	465	15.93	420.67	39	0.879	30
42*	150	A	175	557.46	3.01	-43.51	465	22.95	447.45	80	1.522	30
43*	150	R	169	546.09	5.86	-43.62	465	21.60	448.44	82	0.473	30
44	150	R	168	537.46	2.93	-44.94	465	22.68	445.67	83	0.643	30
45	150	R	168	557.46	-2.93	-35.07	465	20.79	447.79	80	1.285	30
54*	200	R	157	1,114.93	14.66	-53.16	500	19.98	481.48	43	1.014	5

+Run conducted at 12,300 rpm
*Gear scored or failure
**T = Tangential A = Axial R = Radial

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The failure mode occurred suddenly with no evident increase of temperature or torque. Further testing at 13.79 x 10^8 Pa (200,000 psi) tooth pressure was curtailed in order to minimize further damage to the test rig and to conserve the rapid depletion of test gears.

The criteria used to determine the optimum mist nozzle position was:

(a) Visual examination of the test gears

(b) Heat generation (Q_{out}) of the test gearbox

Visual examination of the test gears was sufficient to compare the condition of the contact surfaces and was able to be accomplished without the time consuming removal of the gears for bench measurements. Based on the scoring and/or failure data, shown in Table 4 the best nozzle position was radial followed by the tangential and axial positions.

The heat flow criteria (Q_{out}) is shown in Tables 2 and 3 with a graphical representation depicted in Figure 11. The axial mist nozzle position ranked higher than the radial position when comparing the heat flow (Q_{out}) at the 6.895 x 10^8 Pa (100,000 psi) and 10.34 x 10^8 Pa (150,000 psi) tooth pressure when using either of the test lubricants.

The optimum mist nozzle position selected was the radial mode due to the fewer occurences of gear scoring in that position compared to the axial nozzle position. The slight thermal superiority of the axial position compared to the radial position does tend to confirm Exxon's choice (Appendix 2); however, their testing was more limited in both test duration and number of tests conducted.

One obvious anomaly in the test was the air/oil ratio which varied from test to test. Oil flows from the Norgren mist generator were difficult to control precisely. The maximum, medium and minimum oil feed rates were obtained by an integral oil feed screw in the mist generator head and the actual oil feed rates were not known until the end of each test run when the reservoir oil depletion was measured. Exxon Laboratories encountered similar problems with the repeatability of the air/oil ratios when conducting their tests.

The heat balance results produced from the test data did not give satisfactory conclusions. Contributing to this discrepancy was the unaccountable thermal phenomana due to the severe gear failures which occurred. Also contributing was the service oil leaks caused by the prolonged severe running conditions. These leaks combined with the mist air discharging into the test cell produced an oil laden atmosphere which the ventilation system could not absorb. This oil laden atmosphere coated the test rig and ancillary equipment with an oil film which modified the conduction and calibration factors used in determining the heat balance.

TABLE 4 PRELIMINARY TEST RESULTS

GEAR TOOTH SURFACE EXAMINATION

Gear Tooth Contact Pressure	Radial Nozzle Position						Axial Nozzle Position							Tangential Nozzle Position					
$6.895 \times 10^8 P_a$ (100,000 psi)	0	0	0	0	0	0	0	0	0	●	0	0	0	0	0	0	0	0	0
Test Run Number	7R	8	9	33	34	36	4	5	6	7	31	32	33	1	2	3	28	29	30
$10.34 \times 10^8 P_a$ (150,000 psi)	0	0	●	0	0		●	●	●	●	●			●	●	●	0	0	0
Test Run Number	16	17	43	44	45		13	14	15	41	42			10	11	12	37	38	39
$13.79 \times 10^8 P_a$ (200,000 psi)	●	●	●																
Test Run Number	27	27R	54																

0 Test Run

● Test Run with Scored or Failed Gears

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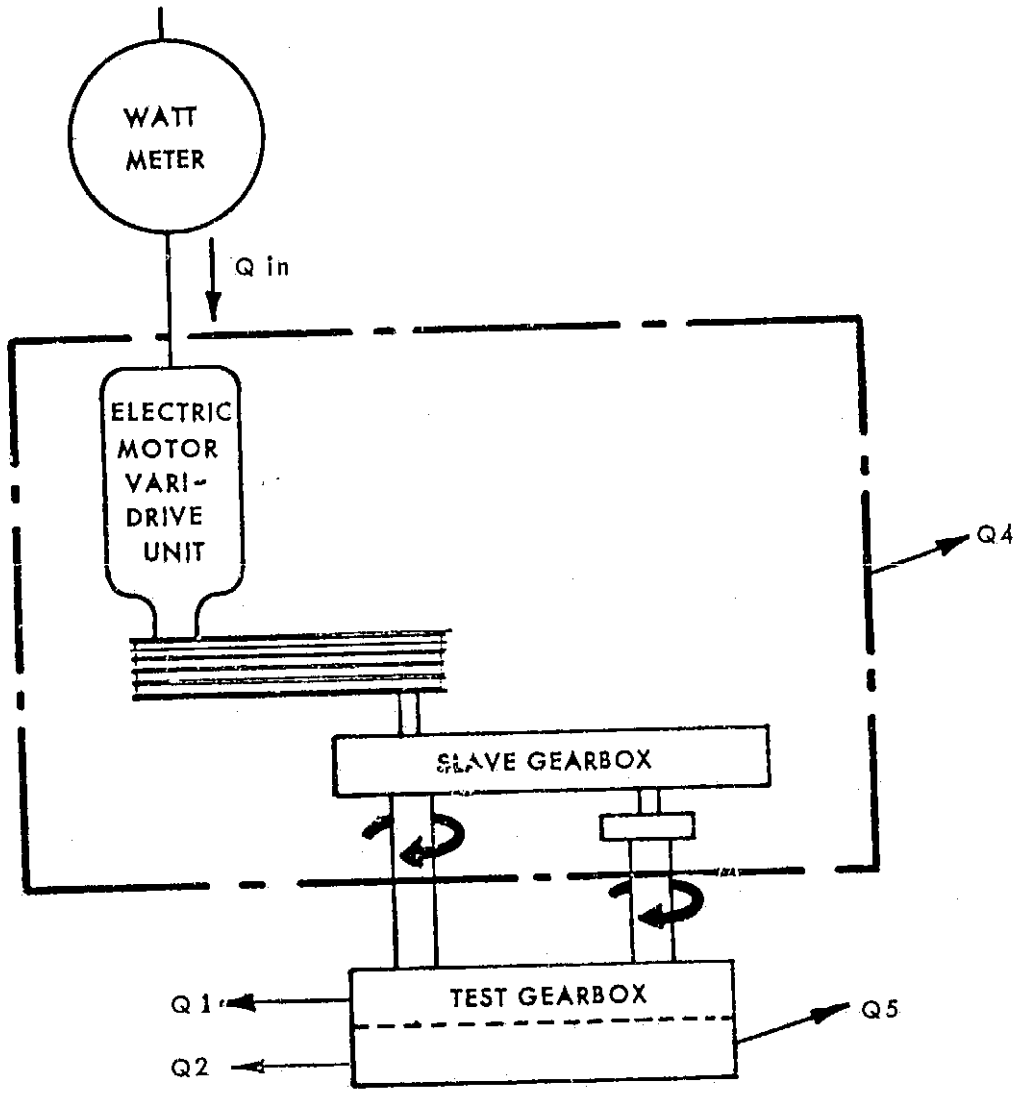

FIGURE 10 HEAT BALANCE SCHEMATIC - PRELIMINARY TESTS AND STEP SPEED TESTS

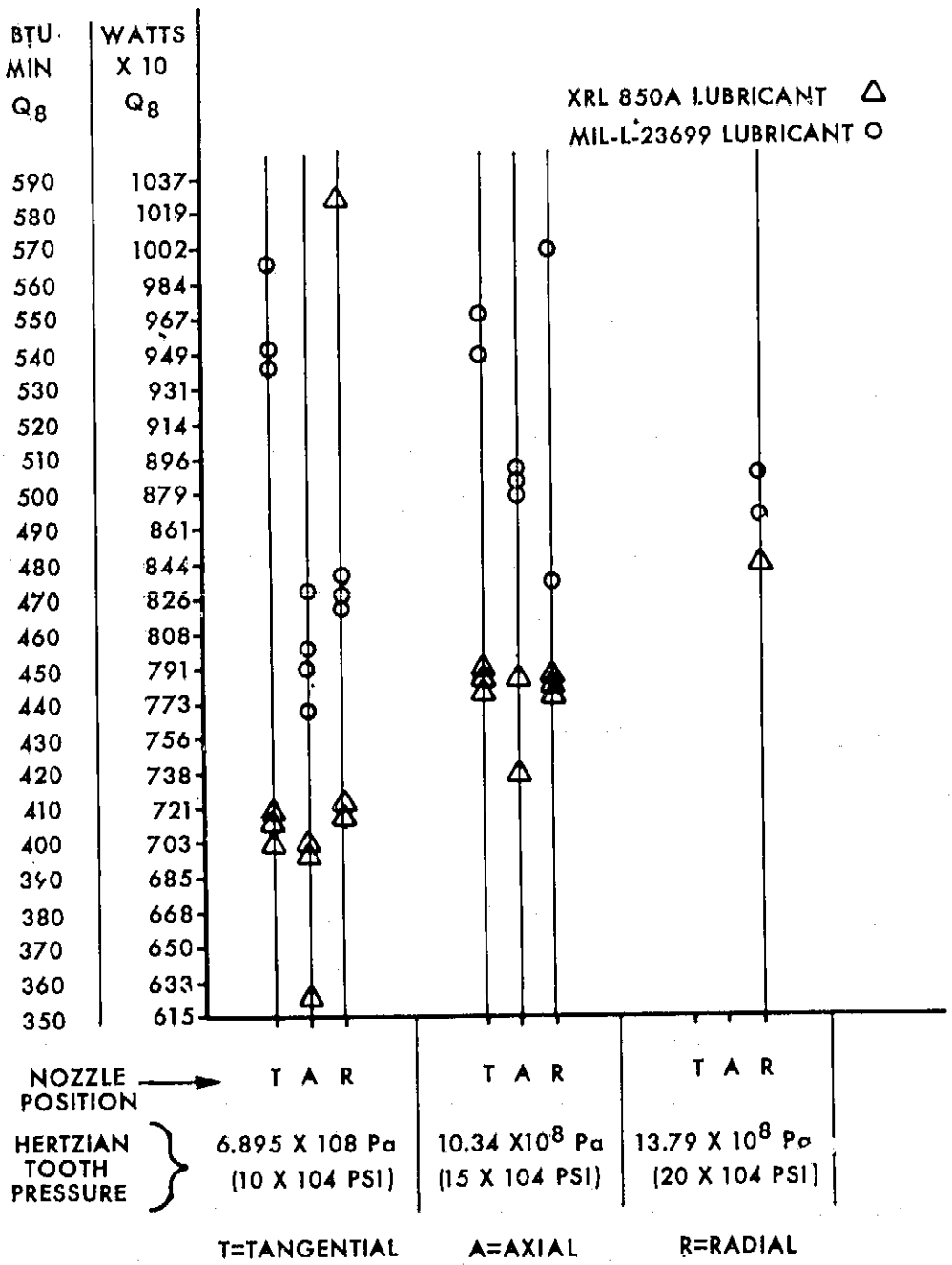

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FIGURE 11 HEAT OUTPUT Q_{out} - NOZZLE POSITIONS PRELIMINARY TESTS

The negative heat flow Q_2 was attributed to the relatively lower test gearbox temperatures compared to the cooling and mist air inlet temperatures of 366K (200°F). Also the short time period of the air impinging on the hot gear teeth compared to the more lengthy time span of the air passing through the cooler test gearbox favored a negative heat flow. The refrigeration effect due to the mist air flowing through the nozzles was found to be minor and discounted (see Appendix 2).

3.1.3 Comparison of Sikorsky Aircraft and Exxon Tests

A survey of the Exxon tests (Appendix 2) revealed two tests which were conducted at speeds, loads, nozzle position and lubricants similar to the Sikorsky mist lubrication tests. The relevant data is presented in Table 5. Comparison of the Exxon and Sikorsky Aircraft data was confusing due to differences in mist/air oil ratios and the cooling air flows, consequently no correlation was attempted.

3.2 Step Speed Mist Lubrication Tests

Using the optimized mist lubrication system, as determined in the preliminary tests previously conducted, a series of step speed tests were performed on the Sikorsky Aircraft two mesh test gearbox.

3.2.1 Test Procedure

Cooling air flow was set at 0.316 mol/s (15 SCFM) per mesh, which was the maximum continuous output available on the test rig. The radial mist nozzle position was selected for both gear meshes with a mist air flow at 0.0632 mol/s (3 SCFM) per mesh and the oil flow (W_4) set at the maximum setting. The actual oil flows are shown on Table 5. Two lubricants were used and the test rig was operated with 10.34×10^8 Pa (150,000 psi) gear tooth contact stress. The gear tester was operated at progressively increasing speeds in increments of 2,000 rpm. Each test was conducted until temperature stabilization (up to 1 hour maximum). A heat balance was established for each condition and the pertinent data is shown in Tables 6A & 6B. A sample calculation is shown in Appendix 1 and a schematic of the heat balance is depicted in Figure 10.

3.2.2 Results and Discussion

Tests conducted with the MIL-L-23699 lubricant produced visual signs of slight scuffing of the test gear tooth contact surfaces at speeds of 10,000, 12,000, 14,000 and 16,000 rpm. Each test was conducted for one (1) hour.

No visible signs of scuffing were observed when using Mobil XRL-850A lubricant at speed of 14,000 rpm and 18,000 rpm. However, slight scuffing was visually detectable at the lower test speeds of 10,000 rpm and 12,000 rpm. Gear tooth failures occurred at speeds of

TABLE 5 COMPARISON OF SIKORSKY AIRCRAFT AND EXXON DATA

Test Run Number	Nozzle Position	T_c Test Gearbox Temperature (K)	Gear Contact Stress P_a	W_4 Oil cc/hr	Cooling Air Flow mol/s	Total Air Flow mol/s	Q_2 H_2* Air Watts
*Exxon 7A	Radial	---	6.90×10^8	19.2	0.545	0.611	119
+Sikorsky 7R	Radial	348	6.895×10^8	3.5	0.316	0.379	-639
+Sikorsky 8	Radial	345	6.895×10^8	9.0	0.316	0.379	-889
+Sikorsky 9	Radial	350	6.895×10^8	13.0	0.316	0.379	-830
*Exxon 11B	Axial	---	10.35×10^8	5.4	0.610	0.674	34
+Sikorsky 13	Axial	364	10.34×10^8	20.0	0.316	0.379	21
+Sikorsky 14	Axial	364	10.34×10^8	31.0	0.316	0.379	22
+Sikorsky 15	Axial	366	10.34×10^8	37.5	0.316	0.379	17

+ Data obtained from Table 2A Preliminary Test Results

* Data obtained from Appendix 2

Tests conducted at 10,000 rpm

Test lubricant MIL-L-23699

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TABLE 6A STEP SPEED TESTS MIST LUBRICATION (SI UNITS)

Gear tooth contact stress 10.34×10^8 Pa
Total mist air per mesh 0.0632 mol/s
Cooling air per mesh 0.316 mol/s
Both mist nozzles in radial position

Speed krpm	Lubricant	Temp. Gearbox Tc (K)	Q_{in} Input (Watts)	Q_1 Service Test Box (Watts)	Q_2 Air (Watts)	Q_4 Leakage Varidrive (Watts)	Q_5 Leakage Test Box (Watts)	Q_{out} Output (Watts)	Gross Balance %	W_4 Oil oz/hr	Test Time (Mins)	Gear Condition
10		353	9,597	464	-720	8,171	394	8,309	87	35	60	Scuffed
12		354	11,795	410	-675	10,016	394	10,146	86	32	60	Scuffed
14	XRL	372	15,194	516	-520	12,828	498	13,322	88	35	60	
16	850A	371	*	348	-447	15,288	484	15,673	--	36	9	Failure
18		367	20,193	55	- 79	18,803	418	19,086	95	36	60	
20		410	*	160	-467	22,844	721	23,873	--	32	36	Failure
10	MIL-	342	9,196	185	-670	8,171	299	7,615	83	33	60	Scuffed
12	L-	350	11,595	371	-505	10,016	323	9,463	82	37	60	Scuffed
14	23699	355	14,394	62	357	12,828	365	12,898	90	36	60	Scuffed
16		363	17,593	192	-366	15,288	408	15,139	86	44	60	Scuffed

* Readings beyond range of wattmeter

TABLE 6B STEP SPEED TEST MIST LUBRICATION (CUSTOMARY UNITS)

Gear tooth contact stress 150,000 psi
Total mist air per mesh 3 SCFM
Cooling air per mesh 15 SCFM
Both mist nozzles in radial position

Speed krpm	Lubricant	Temp. Gearbox Tc (F)	Q_{in} Input (Btu/Min)	Q_1 Serv.Oil Test Box (Btu/Min)	Q_2 Air (Btu/Min)	Q_4 Leakage Varidrive (Btu/Min)	Q_5 Leakage Test Box (Btu/Min)	Q_{out} Output (Btu/Min)	Gross Balance	W_4 Oil oz/hr	Test Time (Mins)	Gear Condition
10		176	546.11	26.38	-40.98	465	22.41	472.81	87	1.18	60	Scuffed
12		179	671.23	23.34	-38.39	570	22.41	577.36	86	1.08	60	Scuffed
14	XRL-	211	864.64	29.35	-29.58	730	28.35	758.12	88	1.18	60	
16	850A	208	*	19.79	-25.45	870	27.54	891.88	--	1.22	9	Failure
18		202	1,149.1	-3.12	- 4.52	1,070	23.76	1,086.12	95	1.22	60	
20		279	*	-9.12	26.60	1,300	41.04	1,358.52	--	1.08	36	Failure
10		157	523.33	-10.55	-38.12	465	17.01	433.34	83	1.12	60	Scuffed
12	MIL-L-	171	659.85	-21.10	-28.75	570	18.36	538.51	82	1.25	60	Scuffed
14	23699	180	819.13	3.52	-20.32	730	20.79	733.99	90	1.22	60	Scuffed
16		194	1,001.16	-10.90	-20.82	870	23.22	861.50	86	1.49	60	Scuffed

*Readings beyond range of wattmeter

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16,000 rpm and 20,000 rpm.

Testing at 20,000 rpm with the Mobil XRL-850A lubricant produced a dramatic failure in the test gearbox which resulted in severe damage to the slave gearbox. Shaft support bearings in both gearboxes were damaged. The gear tooth failure in the test gearbox projected debris through a sight tube containing a calcium fluoride window and into the lens of the infrared thermometer causing extreme damage to the unit. The gear tooth condition of the right side mesh is shown in Figure 12. The left side mesh gear set is shown in Figure 13. The cooling air nozzle which was damaged in this incident is shown at the top center between the two gears. The spray jet shown on the bottom of Figure 13 was inoperative during all the mist lubrication tests. The assembled two mesh gear system is shown in Figure 14. The bulk of the debris generated was placed in front of the open gearbox as shown. In view of the severe damage sustained to the test rig, the remaining testing was limited to speeds up to 16,000 rpm. To preserve the rapidly depleting stock of test gears, previously run gears were used by installing the gears such that the reverse (unloaded) flank of the gear tooth would be subject to the working side of the mesh.

3.3 Jet Spray Lubrication Tests

A series of step speed jet spray tests were conducted in a similar manner as those tests performed in the mist lubrication step speed testing with the exception that a jet recirculating lubrication system serviced the test gear lubrication and cooling.

3.3.1 Test Procedure

The normal jet recirculating lubrication system was installed on the test gearbox, as shown in Figure 1. The gear tester was then operated, using two lubricants. Flow rates are shown in Table 7A (SI units) and Table 7B (Customary units) at increasing speeds, in increments of 2,000 rpm, from 10,000 rpm to 20,000 rpm. Each test was conducted at a tooth load equivalent to 10.34×10^8 Pa contact pressure (150,000 psi) until temperatures stabilized (or 1 hour maximum). Due to the shortage of new test gears, used gears, operating on the reverse unused gear tooth flank, were introduced to the test. A heat balance was established for each condition so a comparison with the mist lubrication step speed tests could be made.

3.3.2 Results and Discussion

Results of the jet spray tests are shown in Tables 7 and a schematic of the heat balance is depicted in Figure 15. Thermal energy flow (Q_{out}) was calculated for each test run and compared to the equivalent mist lubrication test conducted previously. The visual inspection of gears conducted at the end of each test run favored the jet lubrication system, however, a comparison of the thermal energy flow, Q_{out} showed a 15 - 20% increase in heat flow when using the jet lubri-

FIGURE 12 RIGHT SIDE GEAR SET - STEP SPEED TESTING AT 20,000 RPM

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FIGURE 13 LEFT SIDE GEAR SET - STEP SPEED TESTING AT 20,000 rpm

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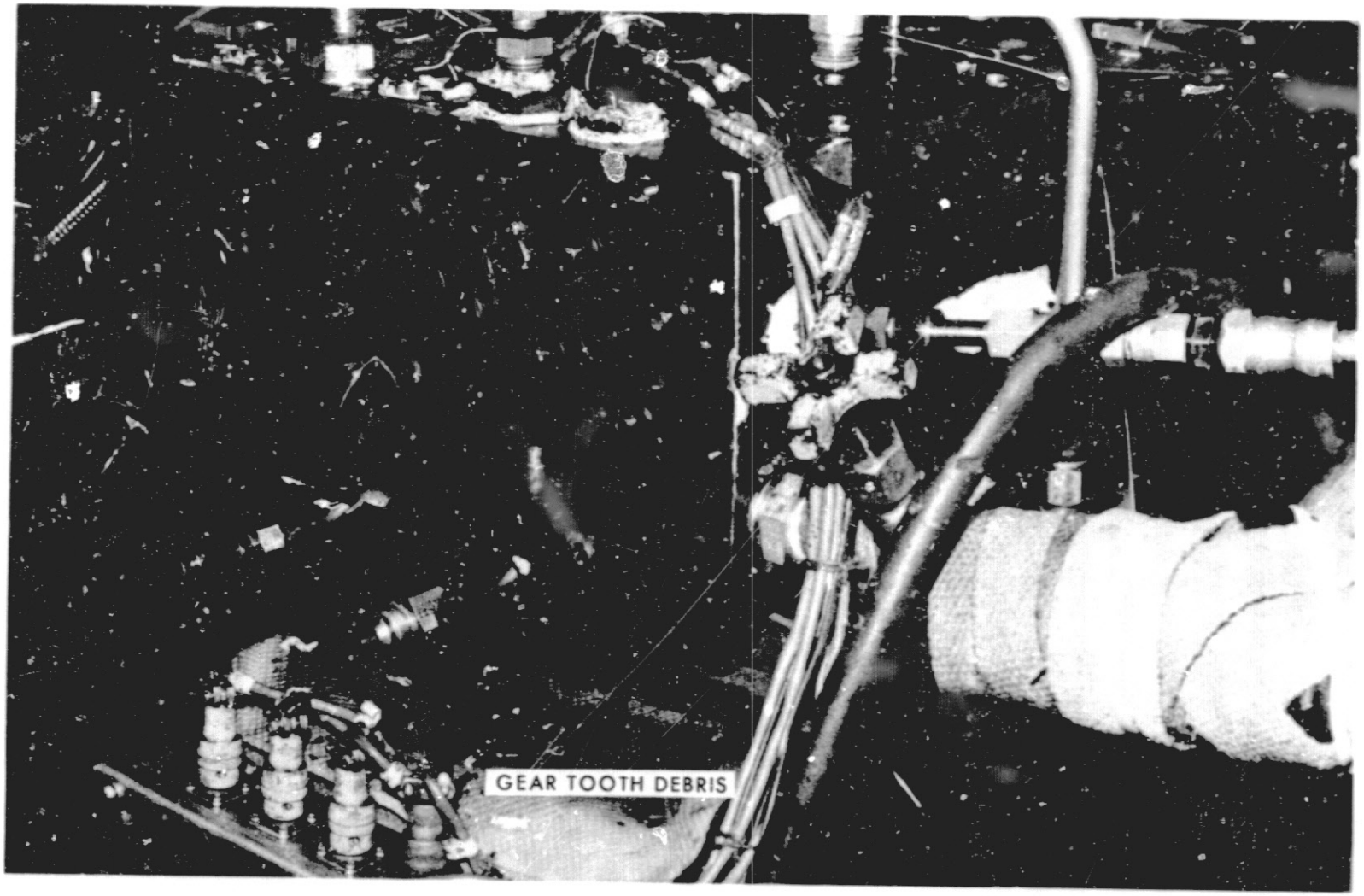

FIGURE 14 TEST,GEARBOX GEAR ASSEMBLY - STEP SPEED TESTING AT 20,000 RPM

TABLE 7A STEP SPEED TEST JET SPRAY (SI UNITS)

Gear Contact Stress 10.34×10^8 Pa

Speed krpm	Lubricant	Temp. Gearbox Tc (K)	Q_{in} Input (Watts)	Q_1 Service Oil Test (Watts)	Q_3 Test Gear Oil Test Box (Watts)	Q_4 Leakage Vari-drive (Watts)	Q_5 Leakage Test (Watts)	Q_{out} Output (Watts)	Gross Balance %	W_5 Oil Kg/ Min	Test Time (Mins)	Gear Condition
10		327	9,796	271	1,305	8,171	152	9,899	101	1.74	60	
12		327	12,995	271	1,579	10,016	218	12,085	93	1.74	30	
14	MIL-L	336	15,394	356	1,923	12,828	242	15,348	99	1.74	25	Excellent
16	23699	345	18,393	536	2,403	15,288	266	18,493	101	1.74	30	
18		352	22,392	499	2,781	18,803	308	22,391	100	1.74	25	
20		361	*	589	2,129	22,844	327	25,887	---	1.74	25	
10		325	9,992	334	889	8,171	190	9,583	96	1.81	25	
12		330	12,396	356	1,159	10,016	209	11,740	95	1.81	25	
14	XRL-850A	338	15,194	400	1,488	12,828	233	14,948	98	2.12	25	Excellent
16		345	18,393	456	1,758	15,288	270	17,772	97	2.12	25	
18		353	21,992	496	2,209	18,803	308	21,816	99	2.12	25	
20		361	*	646	2,750	22,844	342	26,581	--	2.12	25	

*Readings beyond range of wattmeter

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TABLE 7B STEP SPEED TEST JET SPRAY (CUSTOMARY UNITS)

Gear Contact Stress 150,000 psi

Speed krpm	Lubricant	Temp. Gearbox Tc (F)	Q_{in} Input (Btu/Min)	Q_1 Service Oil Test Box (Btu/Min)	Q_3 Test Gear Oil Test Box (Btu/Min)	Q_4 Leakage Varidrive (Btu/Min)	Q_5 Leakage Test Box (Btu/Min)	Q_{out} Output (Btu/Min)	Gross Balance %	W_5 Oil Lb/Min	Test Time (Min)	Gear Condition
10		129	557.48	15.44	74.25	465	8.64	563.33	101	3.831	60	
12		130	739.52	15.44	89.88	570	12.42	687.74	93	3.831	30	
14	MIL-L-	145	876.04	20.24	109.41	730	13.77	873.42	99	3.831	25	Excellent
16	23699	162	1,046.70	30.48	136.77	870	15.12	1,052.37	101	3.831	30	
18		175	1,274.25	28.37	158.26	1,070	17.55	1,274.18	100	3.831	25	
20		191	*	33.53	121.14	1,300	18.63	1,473.13	---	3.831	25	
10		125	568.60	18.98	50.57	465	10.80	545.35	96	4.00	25	
12		135	705.39	20.23	65.96	570	11.88	668.07	95	4.00	25	
14	XRL-	149	864.67	22.76	84.65	730	13.23	850.64	98	4.66	25	Excellent
16	850A	161	1,046.70	25.94	100.04	870	15.39	1,011.37	97	4.66	25	
18		176	1,251.49	28.22	125.69	1,070	17.55	1,241.46	99	4.66	25	
20		191	*	36.75	156.47	1,300	19.44	1,512.66	--	4.66	25	

*Reading beyond range of wattmeter

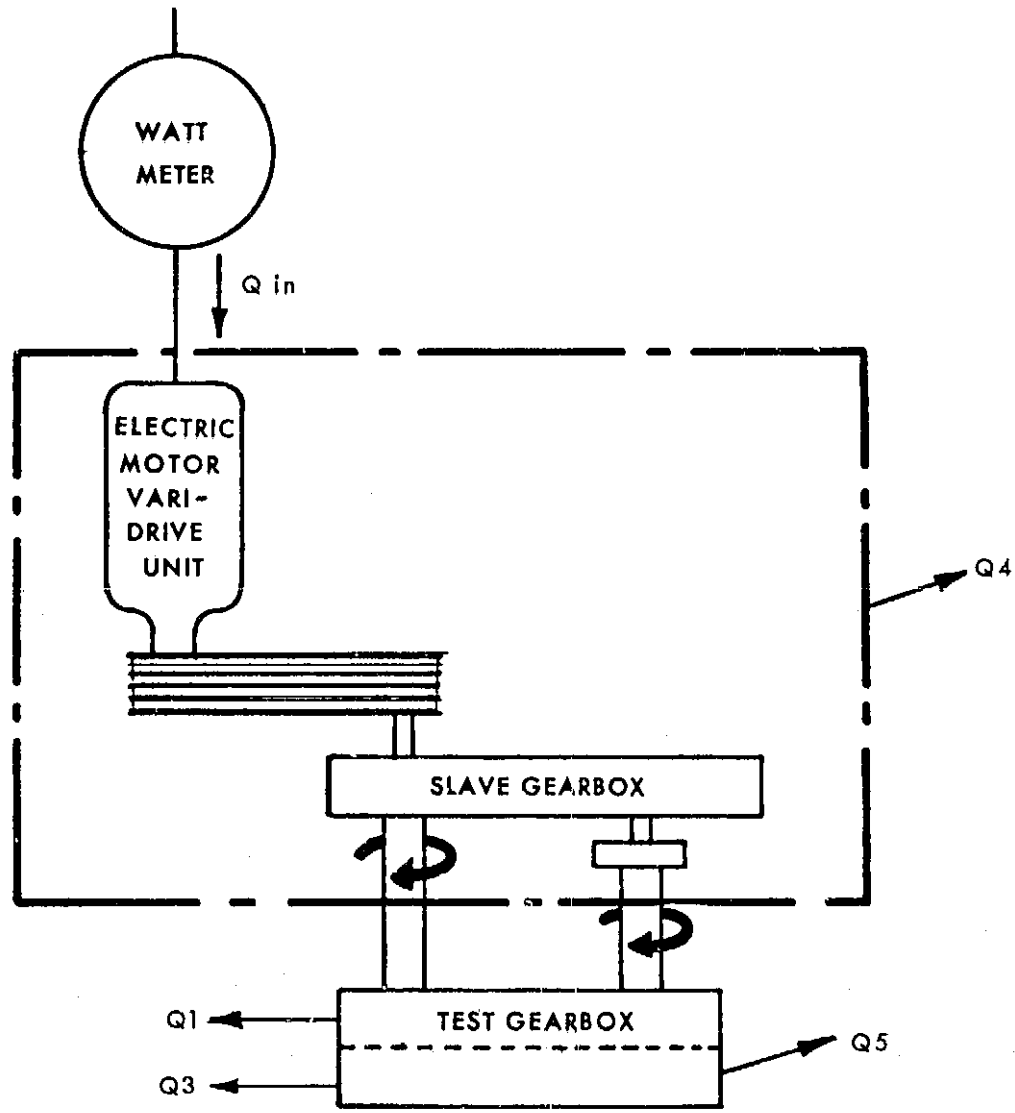

FIGURE 15 HEAT BALANCE SCHEMATIC FOR JET SPRAY TESTS

cation system. A graphical representation of the comparative thermal energy flows is shown in Figure 16.

3.4 Fifty (50) Hour Mist Lubrication Tests

In order to study the gear lubrication effects of prolonged testing using a mist lubrication system, endurance runs using two lubricants were conducted.

3.4.1 Test Procedure

The mist lubricating system was reinstalled on the test gearbox with the mist lubricating nozzles located in the radial position. The test conditions imposed were as follow:

(a) 10.34×10^8 Pa (150,000 psi) gear contact stress

(b) 14,000 rpm

(c) 0.316 mol/s (15 SCFM) cooling air at 366K (200°F) per mesh

(d) 0.0632 mol/s (3 SCFM) of mist air at 366K (200°F) per mesh

An oil feed system was incorporated to increase the apparent limited capacity of the Norgren generator reservoir bowl, which is inadequate for the endurance test runs. The modifications included a manually operated hydraulic pump, positioned outside the test cell, which replenished the reservoir by feeding oil to the reservoir bowl through a port in the generator head, see Figure 17. Oil quantity in the reservoir, oil feed lines and pump were measured before and after each test run. The oil replenishment feature allowed the tests to run continuously for six hours or more, per day.

3.4.2 Results and Discussion

At the end of 27 hours, the test box was partially dismantled. The gears were visually examined and found to be in good condition. Photographs were taken, Figure 18 and the test was successfully continued until completion at 50 hours. The lubricant used for this initial 50 hour test was Mobil XRL 850A and the consumption determined at the end of the test run was calculated to be 22.2 cc/hour per mesh. This test demonstrated the feasibility of using a once through oil-air mist lubrication system in a two mesh gearbox as a replacement for the more vulnerable and heavier recirculating jet spray lubrication system.

The second 50 hour test was conducted under identical conditions as described above with the exception of the test lubricant which was MIL-L-23699. Used gears, reversed in order to use the unused gear tooth flank, were installed. Testing was halted after 90 minutes of running due to rapid rise in load and temperature. The test box was dismantled and examination revealed that the tear teeth were in

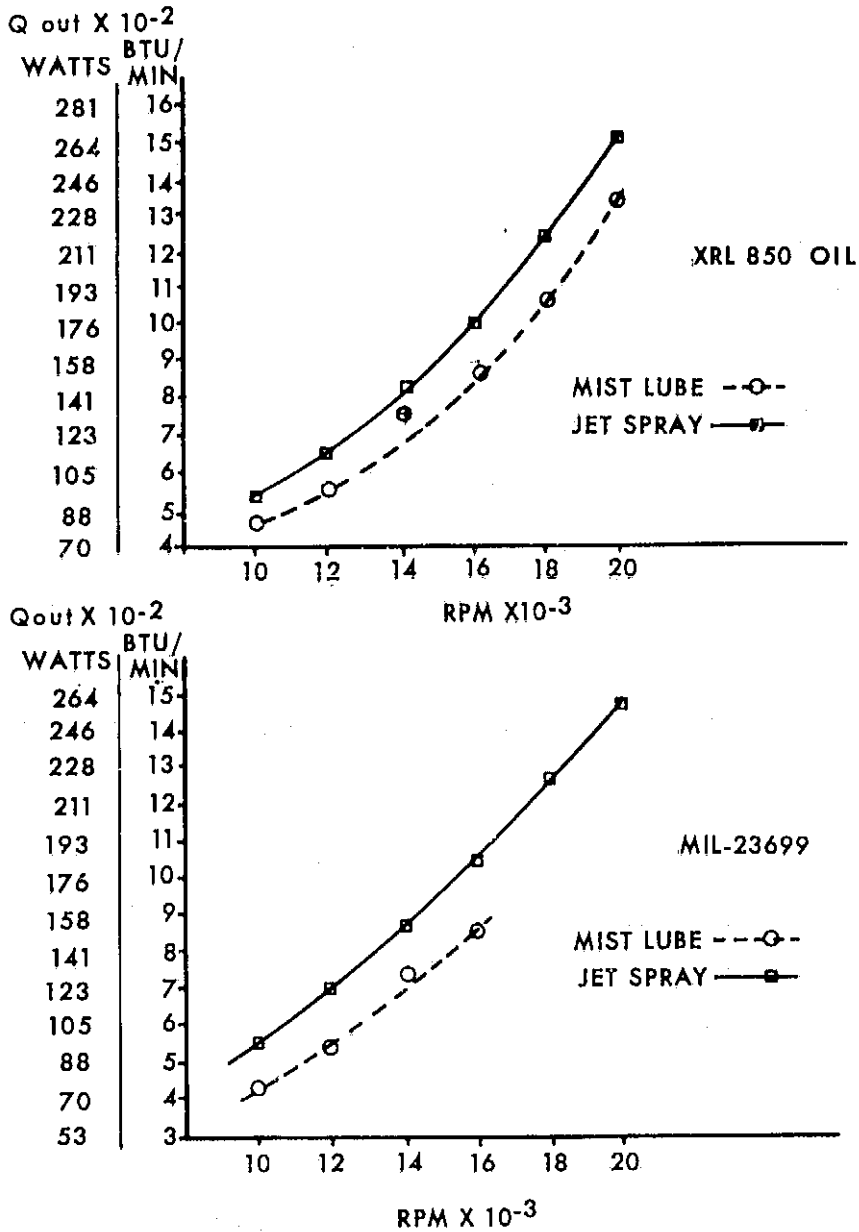

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FIGURE 16 HEAT FLOW Q_{out} FOR MIST LUBRICATION AND JET SPRAY TESTS

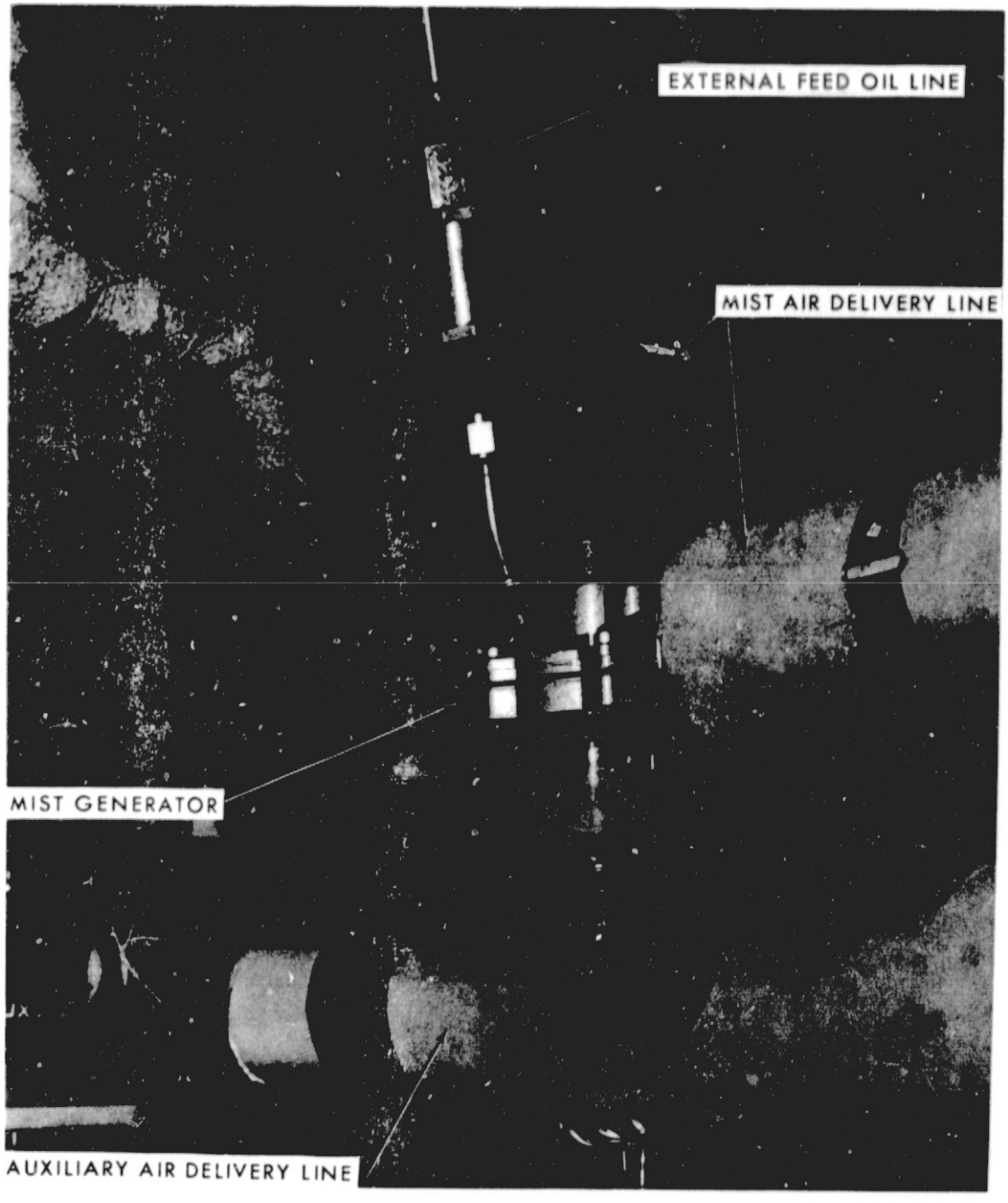

FIGURE 17 EXTERNAL OIL FEED FOR MIST GENERATOR

FIGURE 18 MIST LUBRICATION ENDURANCE TEST 27 HOUR INSPECTION

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poor condition with some tooth breakage. The previously used gears tested may have influenced the limited duration run attained.

3.5 Emergency Aspirator Mist Lubrication System Tests

To demonstrate the practicability of using an aspirator type misting device to serve as an emergency or backup system for conventional or mist lubrication, an aspirating system was designed. The aspirating system was designed to be independent of the mist lubricating and the normal recirculating jet spray system. A simulated local drainage filled lubricant reservoir was placed below the test box to assimilate the approximate location of an emergency reservoir on an aircraft gearbox.

3.5.1 Test Procedure

The emergency aspirator mist lubrication system was installed on the test gearbox as schematically shown in Figure 19. The aspirator nozzles were located in the radial position for both gear meshes. Testing was conducted at 10,000, 12,000, 14,000, 16,000 and 18,000 rpm using ambient air and two lubricants. Pretest runs at no load and 6.895×10^8 Pa (100,000 psi) gear tooth contact stress were conducted to determine air and oil flow rates and to confirm the feasibility of the system before proceeding with the NASA test requirement of 10.34×10^8 Pa (150,000 psi). Tests at each condition were conducted until temperatures stabilized (1 hour maximum) or instability occurred. Following the series of step speed runs, the test box was run using MIL-L-23699 lubricant for five (5) hours at a 10.34×10^8 Pa (150,000 psi) gear tooth contact pressure and 14,000 rpm.

3.5.2 Results and Discussion

Due to the acute shortage of usable test gears, a temperature limitation of 427K (310°F) was set for the center bearing located in the service portion of the test gearbox. Prior to the start of the five (5) hour endurance test, inspection of the slave gearbox revealed the gears in an unserviceable condition. With NASA's consent, high contact ratio gears from NASA Contract NAS3-17859 were substituted. These high contact ratio gears were jet spray lubricated in the salve gearbox while the test gears were lubricated by the emergency aspirator device mist in the test gearbox for the designated five (5) hour test. The results of these tests demonstrated that a simple aspirator mist system is a good candidate for an emergency lubrication system and was operable at rather severe conditions for periods up to five hours without failure. Test results and air/oil flows are shown in Table 8.

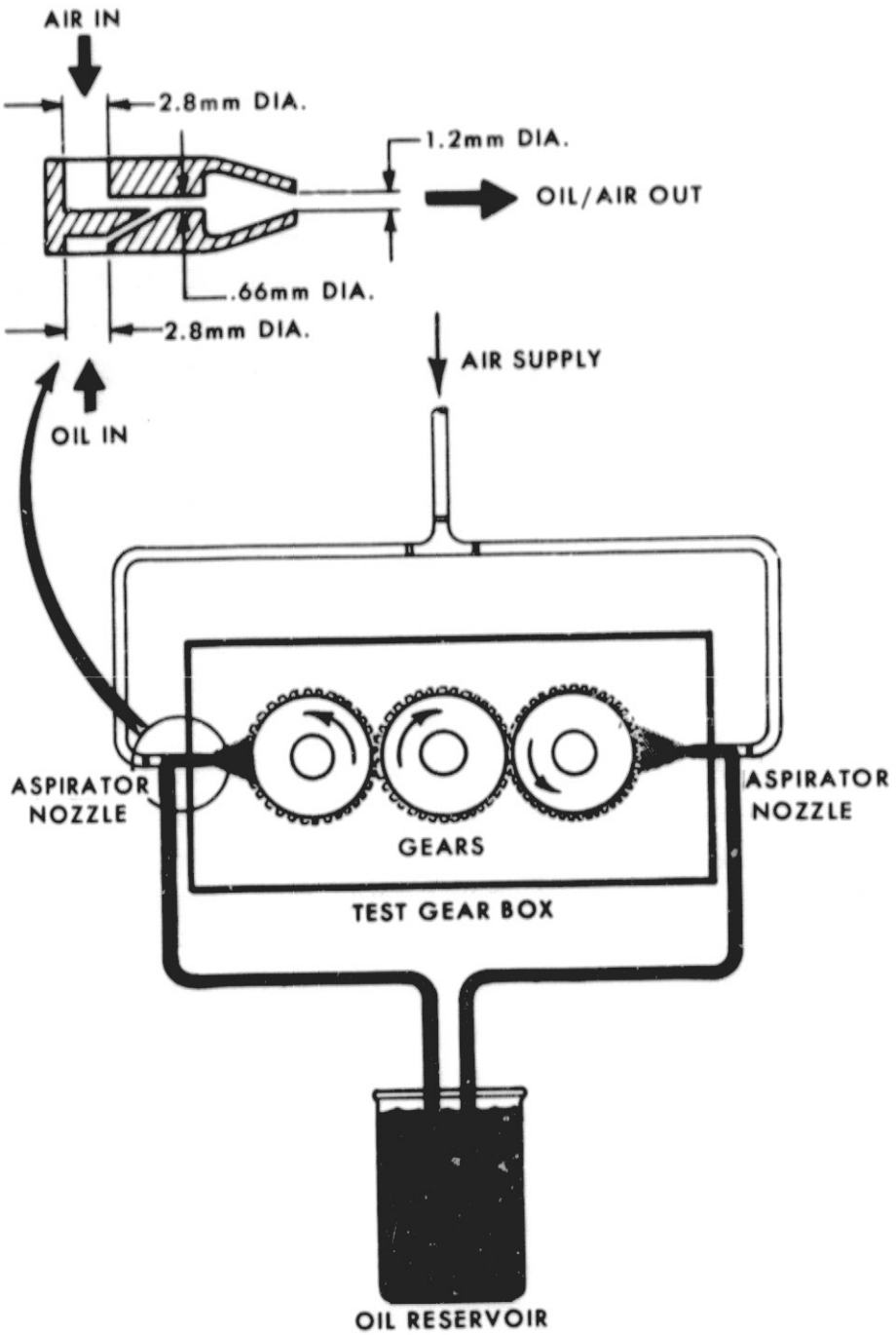

FIGURE 19 ASPIRATOR SCHEMATIC

TABLE 8 EMERGENCY LUBRICATION MIST ASPIRATOR SYSTEM TEST RESULTS

Air flow constant at 0.074 mol/s (3.5 SCFM) per mesh

Lubricant	Gear Tooth Contact Pressure	Speed krpm	Temperature Gearbox T_c	Oil Flow per Mesh cc/min	T_2 Center Bearing	Test Duration Minutes
XRL-850A	6.895 Pa x 10^8 (10×10^4 psi)	10	317K (111°F)	3.50	380K (225°)	30
		12	390K (242°F)	3.50	399K (258°)	45
		14	396K (254°F)	2.50	404K (257°)	30
		16	381K (227°F)	2.88	411K (280°)	40
		18	415K (287°F)	3.15	427K (310°)	23
	10.34 Pa x 10^8 (15×10^4 psi)	10	351K (172°F)	3.25	355K (180°)	20
		12	361K (190°F)	1.85	369K (205°)	35
		14	379K (223°F)	2.15	385K (233°)	30
		16	403K (166°F)	1.25	412K (282°)	40
		18	416K (290°F)	1.80	427K (310°)	22
MIL-L 23699	10.34 Pa x 10^8 (15×10^4 psi)	10	346K (164°F)	1.50	352K (175°)	20
		12	360K (189°F)	1.38	369K (205°)	30
		14	397K (255°F)	1.50	408K (275°)	30
		16	414K (286°F)	1.75	427K (310°)	20
		18	411K (281°F)	2.50	427K (310°)	10
		14	367K (201°F)	1.52	369K (205°)	300

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4.0 SUMMARY OF RESULTS

The purpose of this research effort was to evaluate the performance and heat transfer characteristics of a mist lubrication system for a two mesh gearbox containing 10.16 cm (4 inch) diameter gears, operating under varying loads at speeds between 10,000 and 20,000 rpm. The effects of nozzle arrangement, lubricant/air flow ratio and lubricant on gear performance and heat generation was investigated. The following summarizes the results of this effort:

(a) The fifty (50) hour endurance test using a formulated synthetic hydrocarbon (Mobil XRL-850A) lubricant with a gear tooth contact stress of 10.34×10^8 Pa (150,000 psi) running at a speed of 14,000 rpm was successfully completed. A similar test using MIL-L-23699 type II ester was halted after 90 minutes. Previously used gears were tested in the latter endurance test and this may have been a factor in the inferior performance of the MIL-L-23699 lubricant. The former test demonstrated the feasibility of using a once-through oil-air mist lubrication system in a two mesh gearbox as a replacement for the more vulnerable and heavier recirculating jet spray lubrication system.

(b) Limits established with the restricted capacity mist lubrication system employed were based on visual examination of the gear teeth. Cooling air flow was set at 0.316 mol/s (15 SCFM) per mesh and a mist air flow of 0.0632 mol/s (3 SCFM). With a gear tooth contact pressure of 10.34×10^8 Pa (150,000 psi), a speed limit of 14,000 rpm associated with a minimum oil flow of 0.37 cc per minute per mesh was established, using a formulated synthetic hydrocarbon lubricant. Test gears used in determining these criteria were subjected to more than one test; consequently test results may be biased by the previous tests. Further testing would be required to obtain statistical verification.

(c) The emergency aspirator mist lubrication system five hour endurance test was completed successfully using 0.074 mol/s (3.5 SCFM) of air per mesh and 1.52 cc/min of MIL-L-23699 lubricant, at gear tooth contact pressures of 10.34×10^8 Pa (150,000 psi) and a speed of 14,000 rpm. The reservoir was placed below the test gearbox to simulate the approximate location of a small emergency reservoir on an aircraft gearbox. The success of this rather severe test demonstrated than an aspirator mist system is a good candidate for an emergency lubrication system in helicopter transmissions.

(d) The radial nozzle position was selected as the optimum position based on the visual examination of the gear tooth surface. Thermally the axial position appeared to be superior; thus, confirming the recommendations of Exxon based on their earlier testing. Further testing with all other test conditions constant is required to determine the optimum mist nozzle position.

(e) The recirculating jet spray system was superior to the mist lubrication system based on visual examination of the gear tooth surface condition; however, based on the thermal energy flow (Q_{out}), the mist lubrication system was superior.

(f) The mist lubrication system supplied had limited capacity for the Sikorsky Aircraft two mesh gear system. The maximum cooling air per mesh which the system could attain, 0.316 mol/s (15 SCFM) per mesh and the maximum oil feed for the mist air of 32 - 44 cc/hour appeared to be inadequate to lubricate and cool the gear teeth at tooth contact pressures of 13.79×10^8 Pa (200,000 psi). Further testing at this severe condition will require a larger capacity mist lubrication system.

(g) The infrared thermometer system used was ineffective despite intensive efforts to correlate its readings with samples of known temperature. Consequently the actual gear bulk and gear tooth temperatures were not recorded.

SYMBOLS

Q_{in}	Test rig electrical power input
Q_1	Power losses in test gearbox service lubricant
Q_2	Power loss or gain air (cooling, auxiliary and oil-air mist)
Q_3	P.wer losses in gear jet spray lubricant
Q_4	Power loss in varidrive and slave gearbox
Q_5	Power loss from test gearbox case to room
T_c	Test gearbox average temperature
T_a	Ambient temperature
T_1	Temperature difference service oil in - oil out
T_3	Temperature difference jet spray lubricant in - jet spray lubricant out
T	Temperature difference in - out
ma, aa, co	Subscripts for mist air, auxiliary air, cooling air
W_1	Oil flow - test gearbox service lubricant
W_2	Oil flow - mist air lubricant
W_3	Oil flow - jet spray lubricant
M	Air flow
ρ	Specific weight of standard air
Cp_a	Specific heat of standard air
Cp_o	Specific heat of test lubricant
$\overline{K}$	Thermal conductivity factor for test gearbox

REFERENCES

(1) Shim, J. and Leonardi, S. J., "Microfog Lubricant Application System for Advanced Turbine Engine Components", Task I Report, NASA CR-72291, (Sept. 1967).

(2) Shim, J. and Leonardi, S. J., "Microfog Lubricant Application System for Advanced Turbine Engine Components", Final Report, NASA CR-72489, (Sept. 1968).

(3) Shim, J. and Leonardi, S. J., "Microfog Lubricant Application System for Advanced Turbine Engine Components", Task I and II Report, NASA CR-72743, (May 1970).

(4) Shim, J. and Leonardi, S. J., "Microfog Lubricant Application System for Advanced Turbine Engine Components", Phase II, NASA CR-120843, (Aug. 1971).

(5) Petrucco, R. J. and Leonardi, S. J., "Microfog Lubricant Application System for Advanced Turbine Engine Components - Phase III, NASA CR-121271, (July 1973).

(6) Exxon Co. "Data Book for Designers"

APPENDIX 1

SAMPLE CALCULATION FOR TEST RIG CALIBRATION

The calculation for the test rig calibration was based on an energy balance of the system, that is, power into the system equals power out of the system for steady state conditions.

$$Q_{in} = Q_{out} \qquad \text{Equation 1}$$

Where $Q_{out} = Q_1 + Q_2 + Q_3 + Q_4 + Q_5$

Where $Q_5 = \overline{K} (Tc - Ta)$ and $\overline{K} = 8.53 \frac{\text{Watts}}{K}$ determined by heater tests (Section 2.4.1)

$= 8.53 (38.3 - 23.3)$

$Q_5 = 128$ Watts (7.29 Btu/min)

and $Q_1 = W_c \, Cp_o \, (\Delta T_3)$

The lubricant used in the test rig calibration was MIL-L-23699 with a density of 0.92 g/m^3 at 366K (200^{o}F) and a specific heat of 2134 J/KgK at 366K (0.51 Btu/lb F at 200^{o}F) (Reference 6)

$Q_1 = 3.475 \times 35.6 \times (13.33)$

$= 1649$ Watts (93.84 Btu/minF)

$Q_2 = 0$ (No air flow in test box)

$Q_3 = 0$ (No jet spray in test box)

$Q_{in} = 9277$ Watts (527.9 Btu/min)

Substituting the values obtained for Q_2, Q_3, Q_4 and Q_{in} into Equation 1 we obtain:

$Q_4 = 9277 - (128 + 1649)$

$= 7500$ Watts (426.77 Btu/min)

SAMPLE CALCULATION FOR PRELIMINARY TEST AND STEP SPEEDS

Equation 1 $Q_{in} = Q_{out}$

Where Q_{in} = 7397 Watts (420.94 Btu/min)

$Q_{out} = Q_1 + Q_2 + Q_3 + Q_4$

Q_4 = 7556 Watts (430 Btu/min) obtained from Figure 7

$Q_5 = \overline{K}\ (Tc - Ta)$

$= 8.53\ (71.6 - 41.6)$

= 256 Watts (14.58 Btu/min)

and $Q_1 = W_3\ Cp_0\ (\Delta T_3)$

$= 2.78 \times 35.6 \times (23.31)$

= 2307 Watts (131.3 Btu/min)

and $Q_2 = Q_{mist\ air} + Q_{aux\ air} + Qcool\ air$

$= \rho\ Cp_a \left[m_1\ (\Delta T_{ma}) + m_2\ (\Delta T_{aa}) + m_3\ (\Delta T_{ca}) \right]$

$= .02746 \times 1004.16 \left[.04211\ (-27.78) + .08422\ (-32.23) + .63162\ (-29.45) \right]$

= - 620 Watts (35.29 Btu/min)

$Q_{out} = Q_1 + Q_2 + Q_3 + Q_4$

$= 7556 + 256 + 2307 - 620$

Q_{out} = 9499 Watts (540.59 Btu/min)

Gross Balance $= 100 \times \frac{\text{Output Power}}{\text{Input Power}}$

$= 100\ \frac{Q_{out}}{Q_{in}}$

$= 100\ \frac{(9499)}{7397}$

= 128.4%

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Note: When using Mobil XRL-850A lubricant the following physical constants

were used:

Cp_o = 2301 J/KgK (0.55 Btu/lbF)

Density = 0.8 g/m^3

SAMPLE CALCULATION FOR JET SPRAY TEST

Q_{in} = 9796 Watts (557.48 Btu/min)

$Q_{out} = Q_1 + Q_2 + Q_3 + Q_4 + Q_5$

Q_4 = 8171 Watts (465 Btu/min) for 10,000 rpm and 10.34 Pa x 10^8 tooth contact stress obtained from Figure 7

$Q_5 = \overline{K}\ (Tc - Ta)$ and $\overline{K}$ = 8.53 Watts/K (0.25 Btu/minF)

Q_5 = 8.53 (326.89 - 309.11)

= 152 Watts (8.64 Btu/min)

$Q_1 = W_3\ Cp_o\ (\Delta T_3)$

= 2.74 x 35.6 x (2.78)

= 271 Watts (15.44 Btu/min)

Q_2 = 0 (No air flow in jet spray test)

$Q_3 = W\ Cp_o\ (\Delta T_5)$

= 1.737 x 35.6 x (21.11)

= 1305 Watts (74.25 Btu/min)

Q_{out} = 8171 + 152 + 271 + 1305

= 9899 Watts (563.3 Btu/min)

$$\text{Gross Balance} = 100\ \frac{(\text{Output Power})}{(\text{Input Power})}$$

$$= 100\ \frac{(9899)}{(9796)} = 101\%$$

APPENDIX 2

MICROFOG LUBRICATION FOR HELICOPTER GEARING

(NASA CR-134741)

Exxon Research and Engineering Company

A. Beerbower and R. F. Overhoff

NASA Lewis Research Center

Contract NAS 3-16825

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I. INTRODUCTION

This report covers Task I of NASA Contract NAS 3-16825. This work is closely related to other recent NASA studies on Microfog Lubricant Application Systems, (1, 2, 3, 4, 5) but differs from them in that the tests are conducted on real gears in a dynamic system, under loads and speeds which simulate helicopter operation.

Task I included several stages of preliminary work designed to set up the conditions for a more sophisticated Task II to be conducted at the Sikorsky Aircraft Division of United Technologies Corporation. These stages included:

Analysis of the problem, including feasibility.

Optimization studies.

Step-loading tests.

The analysis included surveying the current literature on gear lubrication in general and microfog lubrication in particular. None of the articles were very helpful, but the best of those found are listed in Section IV.

II. SUMMARY AND CONCLUSIONS

The feasibility of operating the Ryder Gear Tester on microfog lubrication, using the NASA reclassifier nozzle and cooling air at 366 K (200 F), has been demonstrated at loads up to 1380 MPa (200 kpsi) and speeds up to 12,500 rpm.

The cooling air nozzle arrangement is considered to be optimized, even though only supply at the demesh point was tested, since injecting the air at any other point would permit loss of temperature by conduction to the gear body. The reclassifier nozzle position was optimized experimentally and the axial position selected over radial or tangential on the basis of wear rate. The cooling air flow rate was optimized at between 0.254 and 0.358 mol/s (12 to 17 SCFM) at 10 krpm, and at about 0.316 mol/s (15 SCFM) at 12.5 krpm, both at 1380 MPa Hertz (200 kpsi). Consumption of mist oil was not completely optimized, but was found to be between 7.8 and 5.4 cm^3/hr at 10 krpm and less than 7.5 cm^3/hr at 12.5 krpm. All tests were made on MIL-L-23699 oil.

Heat balances were not satisfactory, though from 88 to 98% of the input energy was accounted for in the output heat. The problem encountered was that most of the energy was carried away by the service oil and apparently was largely contributed by the bearing circuit. The rather small contribution of the test gears was almost lost in this high background. For this reason, the heat transfer coefficient shows erratic variations not related to operating variables when calculated from the net heat flux (Input minus Service Oil and Leakage).

Coefficients calculated from the heat absorbed by the air streams (cooling plus mist plus auxiliary air) were no better. Even using a "design" heat generation model did not produce an entirely rational pattern, but at least the expected dependence on rpm was detectable. Evidently there was some error in one or more temperature measurements, despite repeated calibrations of the infrared sensors. There was also the suspicious circumstance that the tooth temperature (T_s) was generally lower than the bulk temperature (T_b), and that the last calibration implied emissivity values greater than 1.00 at some temperatures.

Future operations should be free from the high bearing friction problem, which has been traced to poor bearing design in the Erdco Ryder machine. The bearings have circumferential grooves, contrary to all accepted practices. On the other hand, the infrared thermometers will require intensive precautionary measures to improve the quality of the data in all future work.

III. DETAILED REPORT

1. Materials

1.1 The only oil used during these tests (except for the service oil recirculated in the system) was Exxon Turbo-oil 2380, conforming to MIL-L-23699. However, two other oils were acquired for potential use by Sikorsky in Task II. These were a formulated synthetic hydrocarbon used in a previous contract (4), Mobil XRL-850A, and a laboratory blend corresponding to Humble 3157, a super-refined mineral oil which is no longer available commercially. Aside from the amount of 2380 used, there are 20 gal. of each of these three oils in storage as NASA property.

1.2 The contract called for gears of modern helicopter design. This ruled out the sets available from Pratt and Whitney via Eppi Precision Products. In addition, the NASA contract officer stipulated that two narrow gears be used in all tests where step-loading did not require a wide mating gear. This "narrow-narrow" pair gives a much more realistic configuration for heat transfer studies. With the help of the Sikorsky Aircraft Design and Development engineers, the gears shown in Figures A-1 and A-2 were designed. The required specimens were procured from Clipper Industries, and at the same time Sikorsky had gears made from the same heat metal, forged in the same shop, for use in Task II. The Exxon Research and Engineering procurement was 9 wide and 24 narrow gears. The lead check, spacing, profile and red-line charts provided by Clipper with each narrow gears were inspected at Sikorsky and found acceptable. These two are included in the 5 wide and 12 narrow unused gears in storage as NASA property awaiting shipping advice.

2. Apparatus

2.1 The basic equipment used was the Erdco Ryder Gear Tester described in FTMS 791b Method 6508.1 and in ASTM D 1947. This device was developed in the late 1940's to simulate the gearing used in jet aircraft for power take-off. It consists of a 4-square gearbox, in which the load is applied by hydraulic pressure which causes one of the helical slave gears to slide along the other. As a result of this detail, the 0.635cm (0.25 in) wide test gear on the same shaft slides the same distance. The mating gear has a 2.54cm (1.00 in) wide face to avoid loss of mesh. The test gears have been produced by the Pratt and Whitney Division of United Technologies Corporation ever since the machine was developed there. However, they turned over manufacture of the machine to Erdco in the early 1950's.

In addition to providing the hydraulic loading, oil from the same sump is used to lubricate the six bearings. These are of very peculiar design. When Erdco took over manufacture, they removed the conventional bronze bushings with axial grooves and substituted steel backed silver bushings with lead-indium flashing, having circumferential grooves. The purpose may have been to provide hydrostatic support and so prevent shaft skewing. The entire oil loop is shown in Figure A-3. The oil used is MIL-L-6082, Grade 1100.

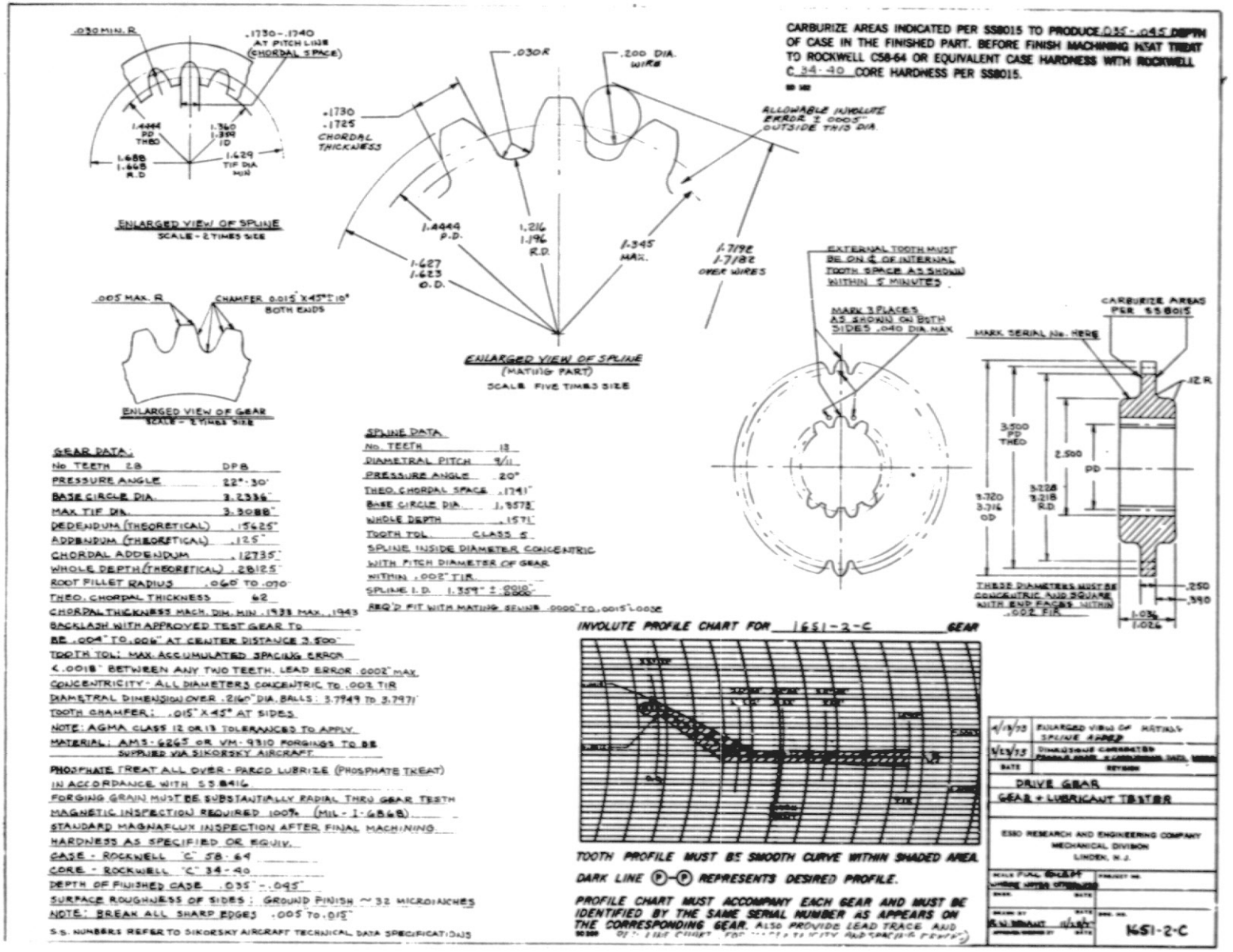

FIGURE 1

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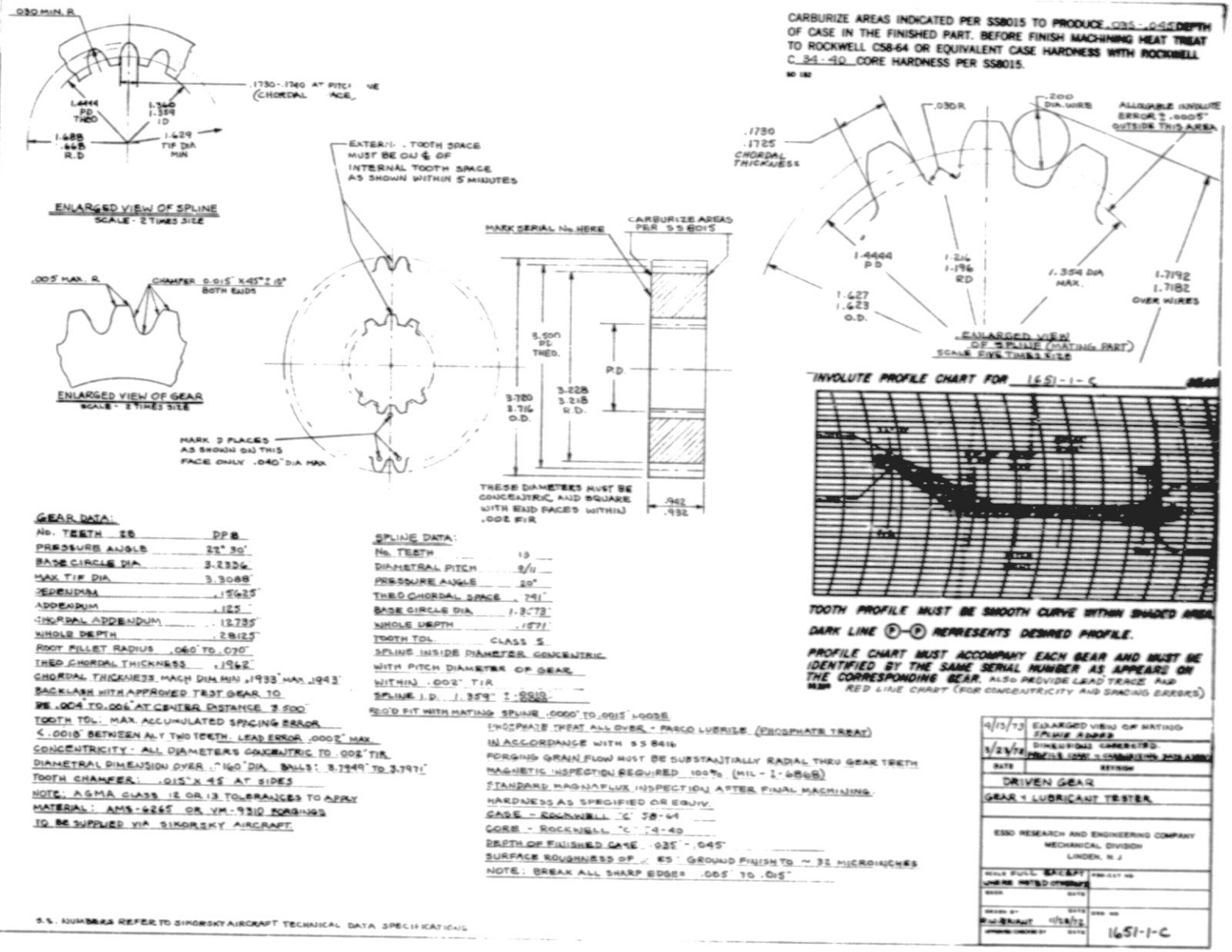


GEAR DATA:

No. TEETH ZG	DP B
PRESSURE ANGLE	22° 30'
BASE CIRCLE DIA	3.2326
MAX. TIP DIA	3.3088"
PERIPHERAL ADDENDUM	.15625"
ADDENDUM	.125"
CHORDAL ADDENDUM	.12735"
WHOLE DEPTH	.28125"
ROOT FILLET RADIUS	.060" TO .070"
THEO CHORDAL THICKNESS	.1962"

CHORDAL THICKNESS MACH DIA MIN .1933" MAX .1943"
BACKLASH WITH APPROVED TEST GEAR TO BE .004 TO .006" AT CENTER DISTANCE 3.500"
TOOTH TOL: MAX. ACCUMULATED SPACING ERROR ≤ .0018" BETWEEN ALL TWO TEETH. LEAD ERROR .0002" MAX.
CONCENTRICITY - ALL DIAMETERS CONCENTRIC TO .002" TIR
DIAMETRAL DIMENSION OVER .160" DIA. BALLS: 3.7949" TO 3.7971"
TOOTH CHAMFER: .015" X 45° AT SIDES
NOTE: AGMA CLASS 12 OR 13 TOLERANCES TO APPLY
MATERIAL: AMS-6265 OR VM-9310 FORGINGS TO BE SUPPLIED VIA SIKORSKY AIRCRAFT.

SPLINE DATA:

No. TEETH	13
DIAMETRAL PITCH	9/11
PRESSURE ANGLE	20°
THEO CHORDAL SPACE	.741"
BASE CIRCLE DIA	1.3.773"
WHOLE DEPTH	.1371"
TOOTH TOL	CLASS 5

SPLINE INSIDE DIAMETER CONCENTRIC WITH PITCH DIAMETER OF GEAR WITHIN .002" TIR
SPLINE I.D. 1.359" ± .0010"
REQ'D FIT WITH MATING SPLINE .0000" TO .0015" LOOSE

PHOSPHATE TREAT ALL OVER - PARCO LUBRITE (PHOSPHATE TREAT)
IN ACCORDANCE WITH SS 8416
FORGING GRAIN FLOW MUST BE SUBSTANTIALLY RADIAL THRU GEAR TEETH
MAGNETIC INSPECTION REQUIRED 100% (MIL - I - 6868)
STANDARD MAGNAFLUX INSPECTION AFTER FINAL MACHINING.
HARDNESS AS SPECIFIED OR EQUIV.
CASE - ROCKWELL "C" 58-64
CORE - ROCKWELL "C" 34-40
DEPTH OF FINISHED CASE .035" - .045"
SURFACE ROUGHNESS OF ALL EXCEPT GROUND FINISH TO ~ 32 MICROINCHES
NOTE: BREAK ALL SHARP EDGES .005" TO .015"

S.S. NUMBERS REFER TO SIKORSKY AIRCRAFT TECHNICAL DATA SPECIFICATIONS

9/13/73	ENLARGED VIEW OF MATING SPLINE ADDED
5/21/73	DIMENSIONS CORRECTED. PROFILE CHART CORRECTED AND ADDED
DATE	REVISION

DRIVEN GEAR
GEAR & LUBRICANT TESTER

ESSO RESEARCH AND ENGINEERING COMPANY
MECHANICAL DIVISION
LINDEN, N. J.

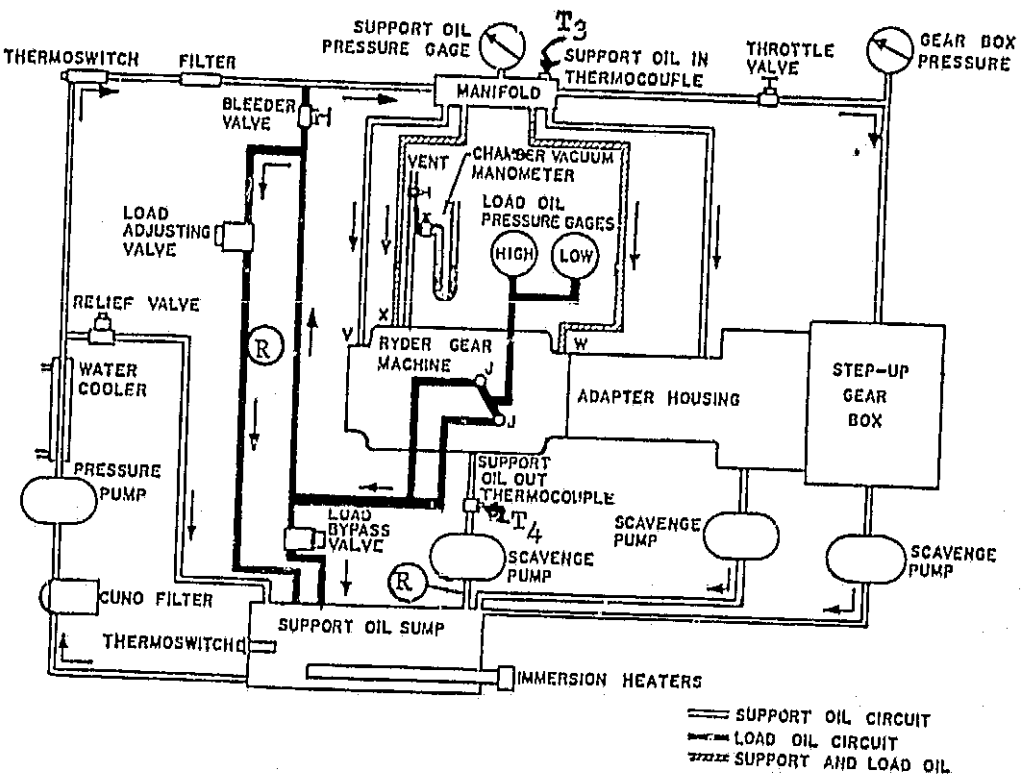
DRAWN BY	DATE 10/18/72
CHECKED BY	DATE
APPROVED BY	DATE

1651-1-C

FIGURE 2

FIGURE A-3

SUPPORT AND LOAD OIL SYSTEMS – ERDCO RYDER GEAR MACHINE



R = point for flow rate measurement

V = support oil to outboard bearings. This line had to be put outside the case and discharged direct to the scavenge pump to avoid interference with T_s.

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2.2 The Ryder gear unit was already fitted with a microfog lubrication and cooling system from pre-proposal work. However, most of this had to be replaced to meet the contractual requirements, and only the final system will be described. The cooling air nozzle is shown in Figure A-4. The microfog generator is Norgren Model No. 10-015-002, the smallest available. It was not equipped for precise measurement of the oil flow, so a Sage Instrument Company infusion pump (Model 234-4) was used to replace the normal aspirator feed action. As part of the initially formed mist is too coarse to be used as microfog, that part is removed in the generator and returned to the sump, for recycle. This was measured and subtracted from the rate programmed on the Sage pump. Thus, the amount of oil actually delivered as microfog was never exactly known until the end of the run.

One detail had to be changed, with the consent of the Project Manager. The contract called for the generator to be held at 366K (200F) but with the modified oil feed the best approach was to preheat the misting and boost air streams to that temperature by heating tapes on the lines and Variac control. Since the plastic bowl supplied with the generator had a safety rating of 100 psig at 200F, and a regulator malfunction could apply the full 100 psig line pressure to it, a heavy glass bowl was fabricated by our glassblower.

2.3 The reclassifier nozzle used was based on that developed for NASA under previous contract (5) and is shown in Figure A-5. There did not seem to be any point in protecting the gears from drops that might form on the tip, so the air sheath in the original design was omitted. However, it was considered desirable to use the existing auxiliary "boost" air rotameter to maintian the nozzle flow rate in the design range of 0.034 to 0.088 mols/s (1.63 to 4.20 SCFM) regardless of the generator "mist" flow. In practice, total flow was maintained at 2.5 to 3.0 SCFM, of which 45±5% was mist air. It was realized near the end that this resulted in oil/mist air ratios which probably could not be obtained in an aspirating generator, so the last run (MF-13) was made with 0.9 SCFM of mist and 1.1 SCFM of boost air.

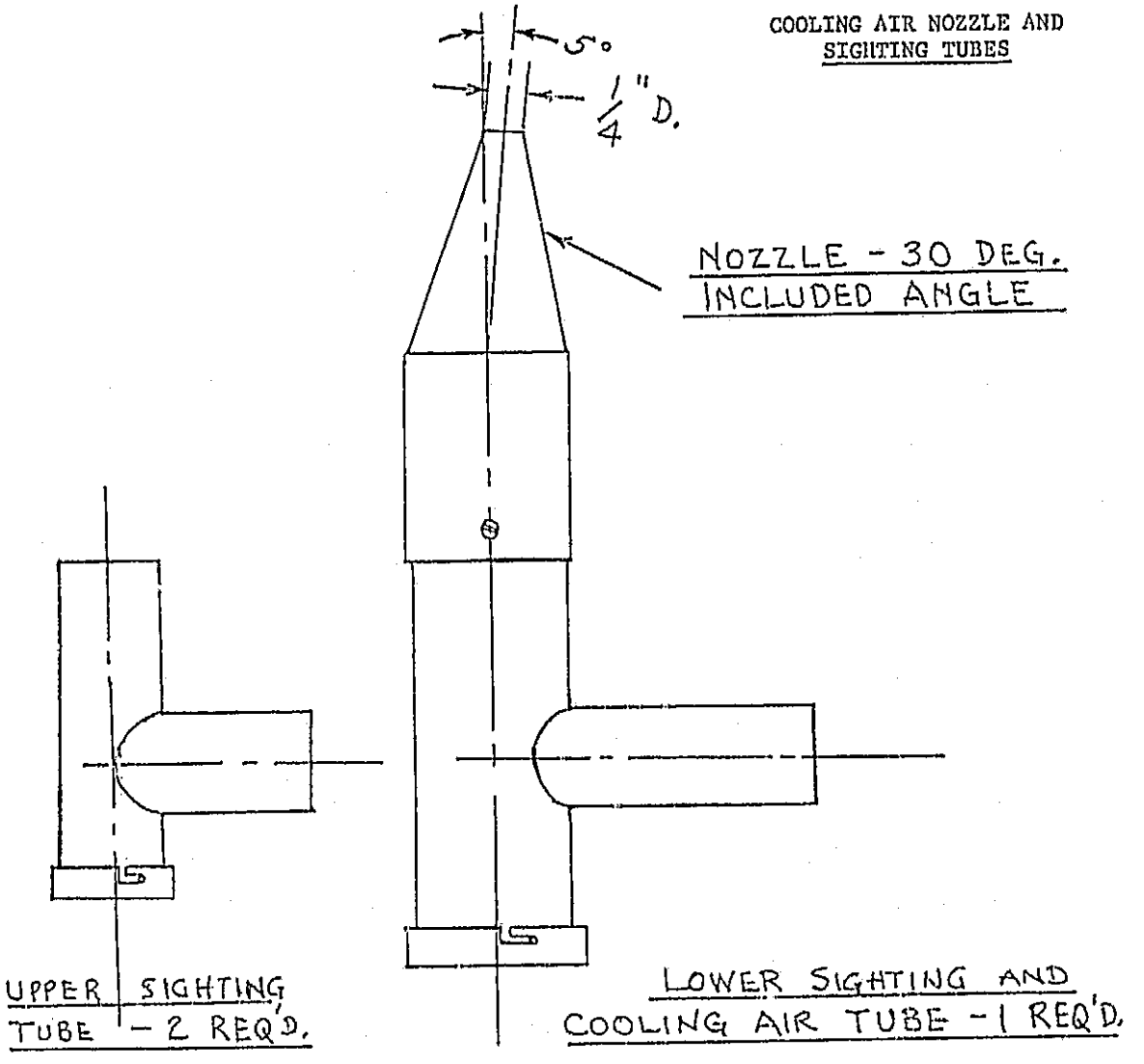
2.4 The contract called for the cooling air to be at 366K (200F). This was arranged by passing house air over a 2000 watt vayonet heater, Chromalox Model MLO-215AW, in a section of pipe. This was controlled in off/on mode by a Gardsman Model JP. Finer control was achieved by adding a "base load" heater, Chromalox MTO-220A, on Variac control. Air flow rates were measured at room temperature by existing rotameters and pressure gages. Standard charts were used to compute these readings into mols/s and SCFM (cubic feet per minute at standard conditions of 1 atmosphere and 32F).

2.5 Temperatures at most points were measured with existing thermocouples wired to indicators and records in the control room. A schematic of the readings needed for calculations is shown in Figure A-6.

2.6 Tooth and bulk temperatures of the gears were to be measured by infra-red thermometers recently purchased by the contractor. There were Mikron Model 66. While sound in principle, these instruments presented

FIGURE A-4

COOLING AIR NOZZLE AND SIGHTING TUBES

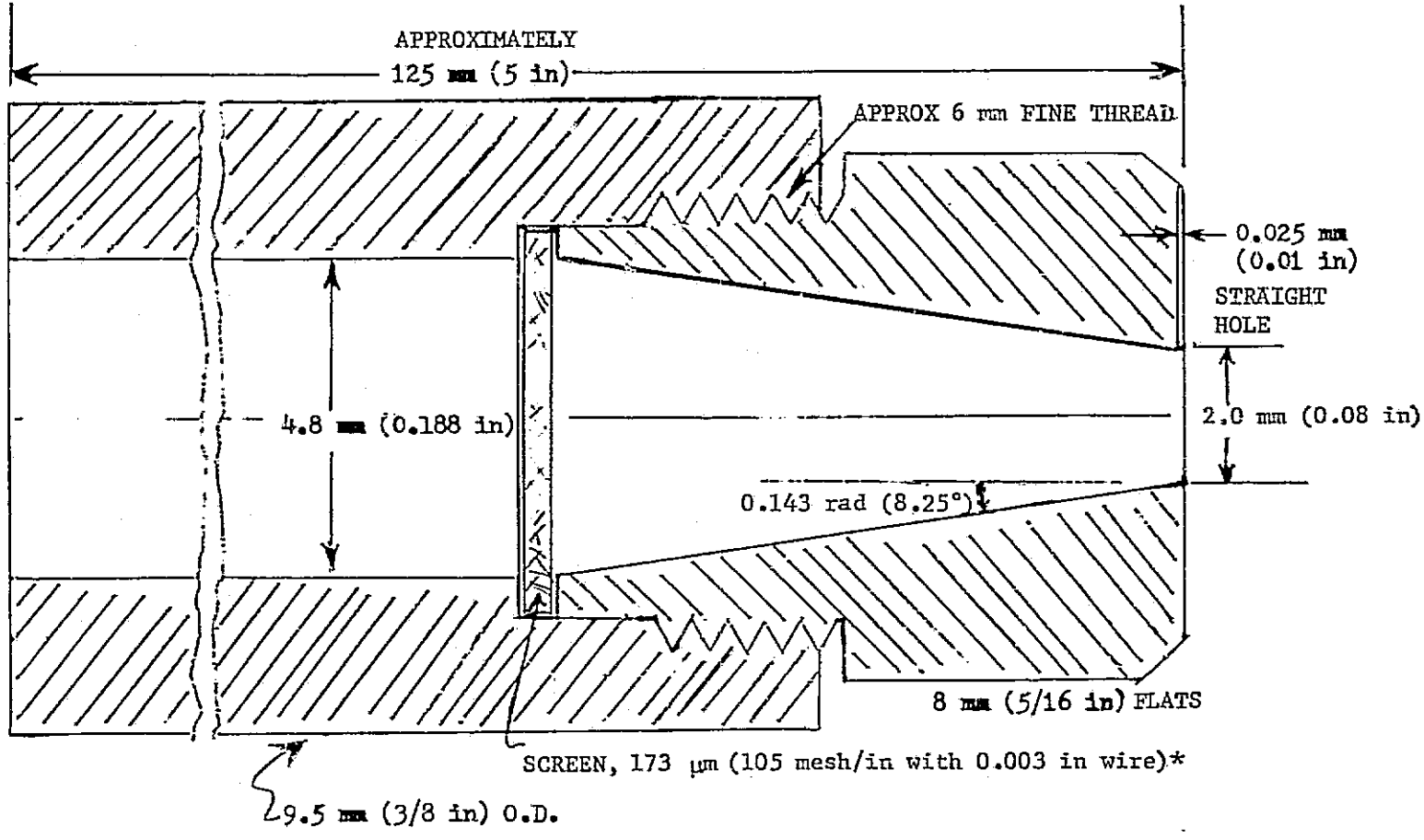


NOTE : LOWER SIGHT TUBE AND COOLING AIR NOZZLE ROTATED TO ALIGN WITH AXIAL POSITION OF GEARS.

FIGURE A-5

DETAIL OF NASA

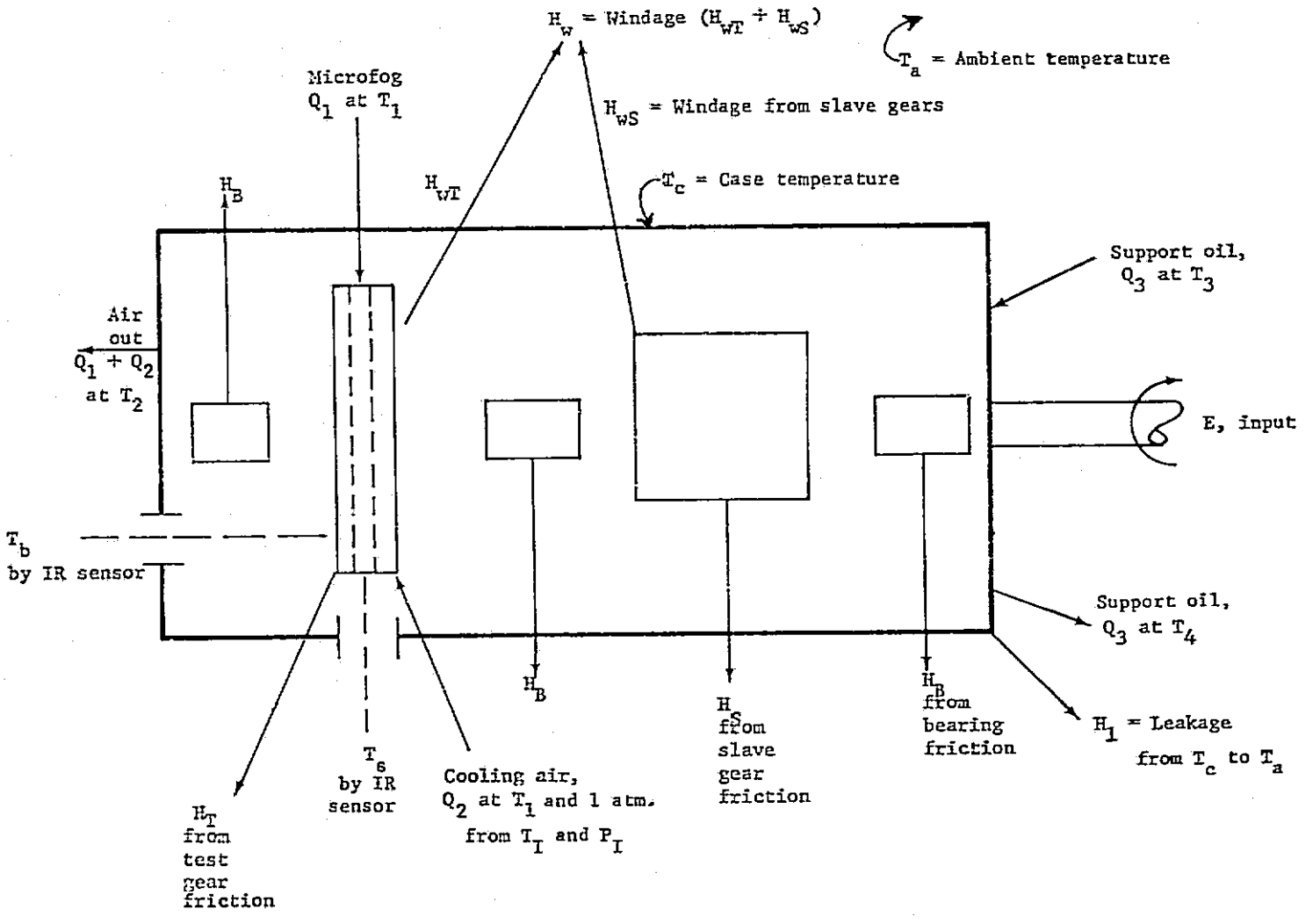
MICROFOG RECLASSIFIER NOZZLE



*Cambridge Wire Cloth Co., 93316-A1050-0030A-24000

FIGURE A-6

SCHEMATIC DIAGRAM OF A GEAR BOX, FOR THE HEAT BALANCE CALCULATIONS



serious calibration problems, as discussed below. Sighting tubes were required, as shown in Figure A-4. The orientation of the tubes on the head is shown in Figures A-7 and A-8.

2.7 Input power was to have been measured by a wattmeter, but the nature of the speed control on the Erdco Ryder test-bed made this impossible. A strain gage system recently purchased by the contractor was used. Since the high shaft speed ruled out slip rings, the Accurex Model 1206 torque telemetry system was selected. Some calibration problems were encountered and are discussed below.

3. Calibration of Equipment

3.1 The first calibration established was the rate of heat leakage from the gearbox to the environment. Three ring-shaped heating elements, Chromalox Type A-2, (500 watts at 120 volts), were hung on the shafts and wired in parallel to a Variac and a watt-hour meter. The room ventilation was arranged just as if a run were in progress. Thermocouples attached to three points on the box were wired to a recorder and the system left overnight to equilibrate, to be sure the static situation did not result in non-equilibrium conditions. The best run consumed 30 kw hrs in 65 hrs, with T_c = 370K (206F) and T_a = 300K (80F). From this

$$H_1 = 6.61\ (T_c - T_a)\ \text{watts/K}$$

Although exact linearity of this equation was not proven, it will be used since the values of T_c and T_a were quite close to the target values for the real runs. A similar test must be made for Task II, which will use an entirely different gearbox.

3.2 The next calibration was travel of the driven shaft during loading. This proved to be quite difficult, due to hysteresis in the support oil seals and was not in fact feasible by direct action. Spacers were prepared on the basis of the best results from feeler gage testing, and some practice runs made using two narrow Pratt and Whitney gears. Examination of the wear scars showed small unworn steps. Using these to estimate the error permitted preparing spacers for each load which gave more exact alignment during the actual test runs. No similar problems will arise on Task II.

3.3 A good deal of time was spent in attempts to calibrate the service oil flow rates. No difficulty was encountered with the load oil bleed-off rate, as shown in Figure A-9. However, the flow rate of the support oil was so great as to render insignificant the contribution of load oil to total flow, and attempts to obtain comparable precision on support oil were frustrated by some unknown factor. However, rates were consistent during each run, and the unexplained variance arose only when there was a stop and start, or a major change in load or speed. Hence, the support oil flow rate was determined during every run, starting with MF-7, and combined with the load oil rate from Figure A-9 for the total service oil flow rate.

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FIGURE A-7

RADIAL ORIENTATION OF COOLING AIR NOZZLE AND SIGHTING TUBES

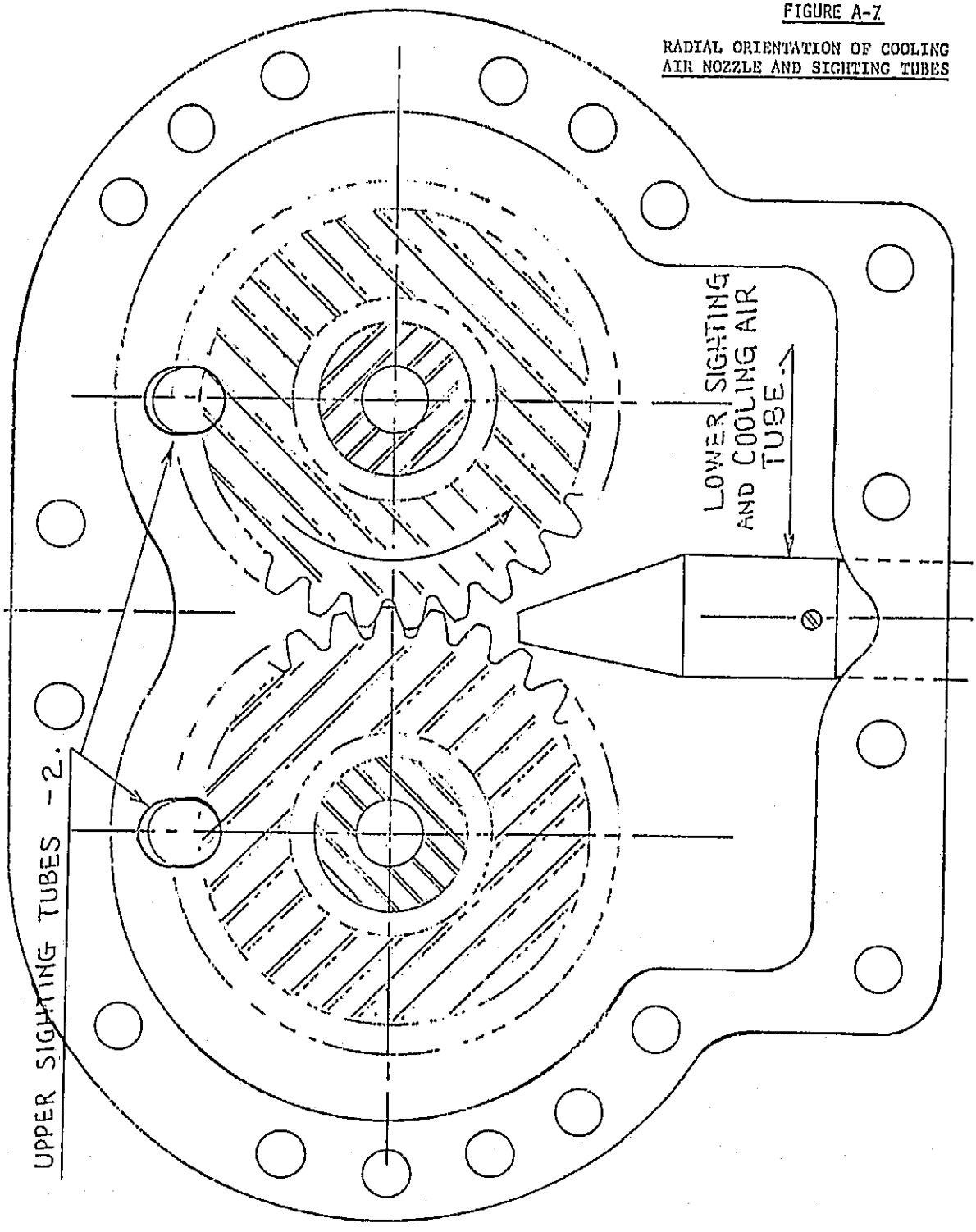

FIGURE A-8

AXIAL ORIENTATION OF COOLING AIR NOZZLE AND SIGHTING TUBES

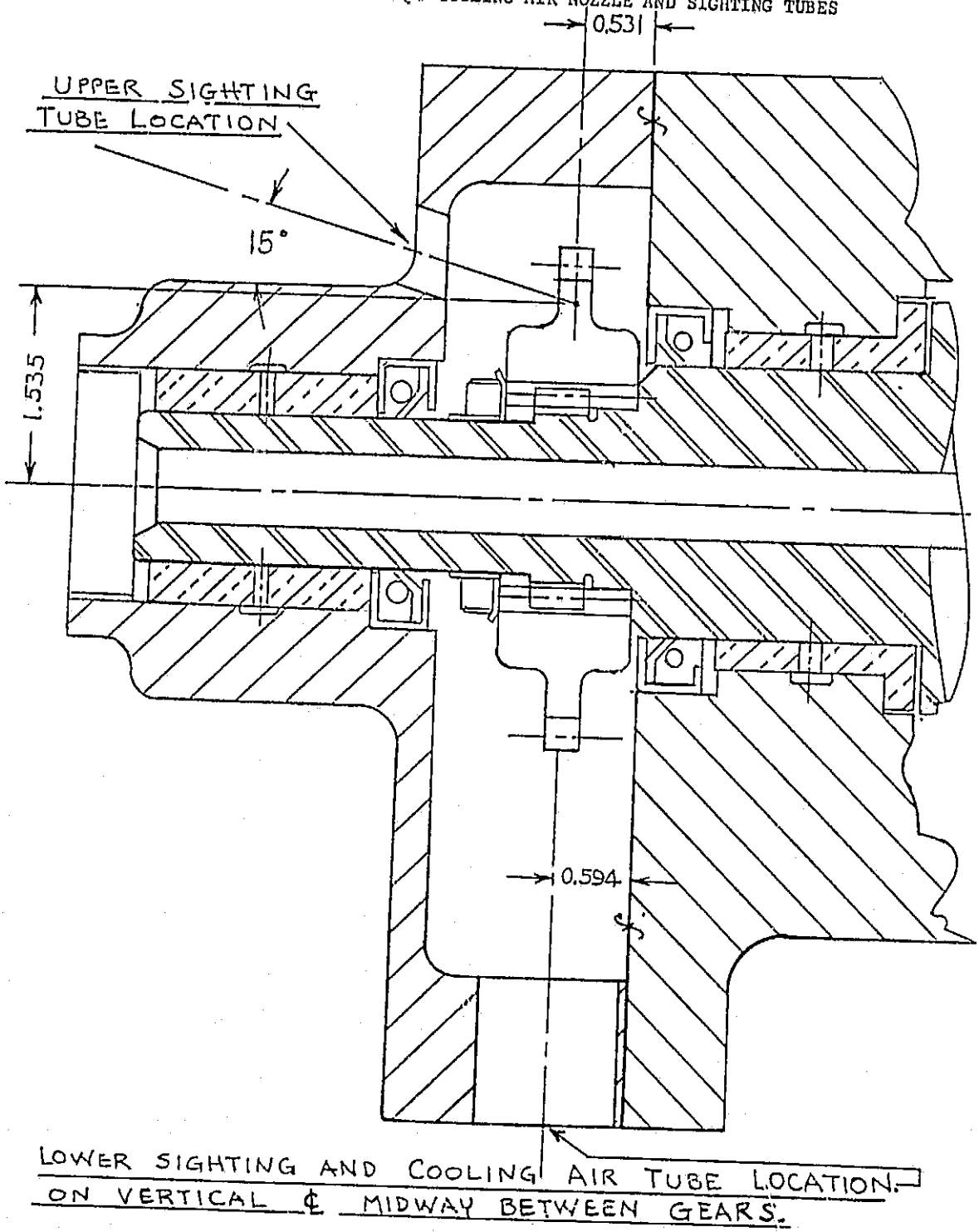

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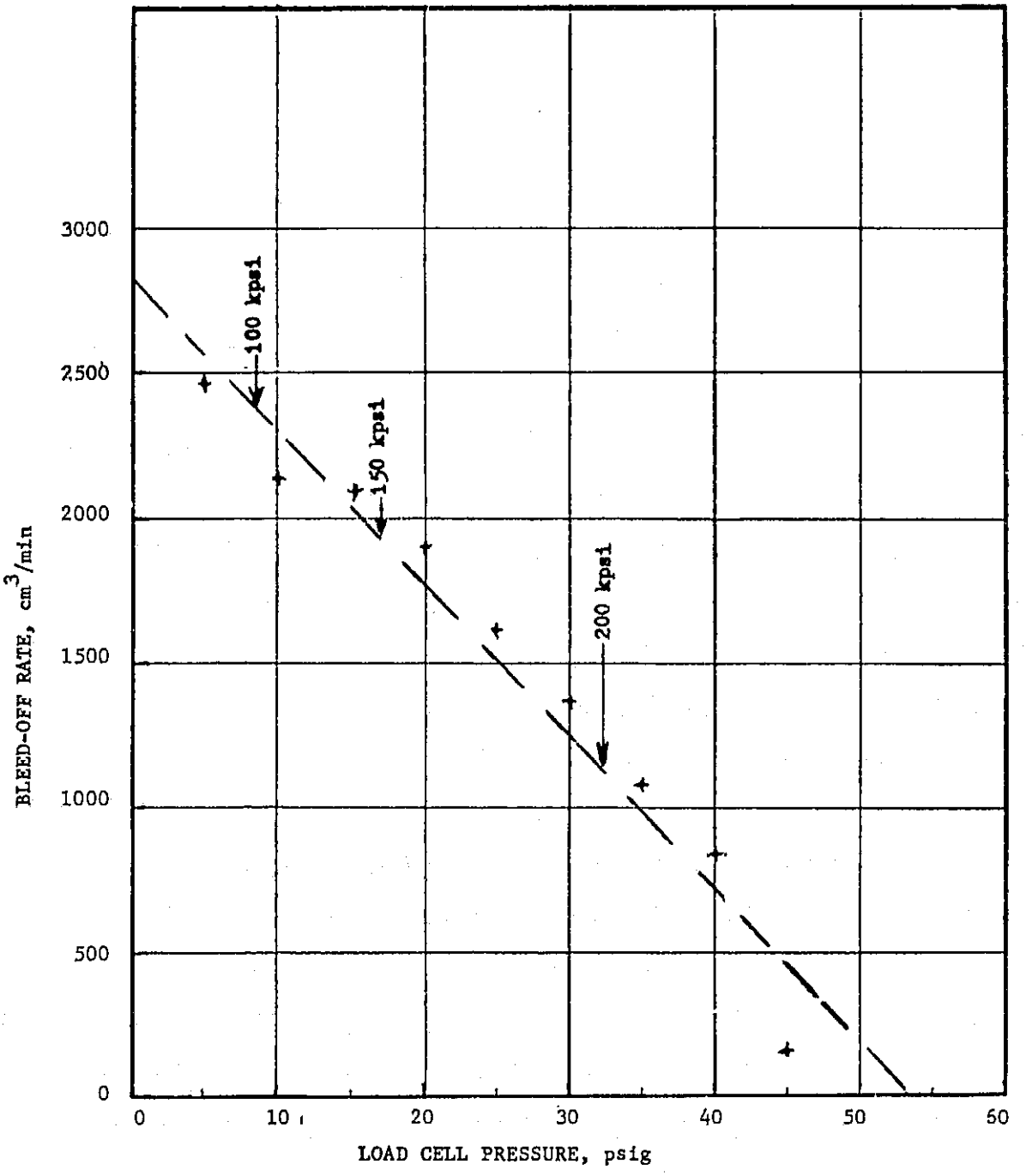
FIGURE A-9

BLEED-OFF RATE OF LOAD OIL VERSUS LOAD



3.4 The Accurex torque measurement system was calibrated by means of a manually operated torque wrench calibrated in lb-in. To establish the integrity of this approach, the calibration was repeated using dead weights on a lever arm and was found to agree closely with the torque-wrench calibration. The hysteresis of the measurements was substantial, due to static friction in the seals used to keep service oil out of the test chamber. The result is that the readings during loading lag about 5% and those during unloading lead by about the same. Assuming that Hooke's Law is obeyed, as is almost always the case in strain gage applications, the lag and lead can be calcelled out as shown in Figure A-10.

Another problem arose quite unexpectedly. After some initial difficulties with loss of bonding of the strain gage elements had been overcome, the system appeared stable. Calibration on December 18, 1973 had given a factor of 0.115 N·m(1.02 lb-in) per division and an accident to the telemetry system on January 17 during Run MF-5 did not appear to have affected this factor. However, at the end of MF-13D, a check was made to verify stability, and the factor was found to be 1.67 as shown in Figure A-10. This gave acceptable heat balances on the last three runs. (MF-13B, C and D) at 12,550 rpm, but appeared to be a little high for all prior work. On the other hand, the first value was acceptable only for MF-5. There had obviously been a claibration shift during repair of the elemetry unit, and perhaps another small shift when the speed was increased. This problem will not be pursued further, as Task II will use the wattmeter rather than the torquemeter.

3.5 The Mikron infra-red thermometers were the most troublesome part of the equipment, for a number of reasons. They proved to be quite fragile, and required several returns to the manufacturer. After each repair, it was necessary to recalibrate. In general, these recalibrations did not retrace the same curves as before repair. Admittedly there were some improvements made in technique of calibration, in addition to whatever changes were made inside the instruments.

The calibration used for most of the runs, starting with MF-8, was made using a P&W gear which had teeth polished with crocus cloth. A thermocouple was welded to it, and the Mikron focused on the appropriate surface. The gear was kept on a hotplate in a can purged with nitrogen to prevent oxide coloring of the polished surface. Calcium floride windows, as used in actual runs to keep mist off the optics, were in the IR path. The temperature was raised in increments, at each of which the emissivity knob was turned to scan from 1.00 to 0.20, the entire range. In theory, the actual and indicated temperatures should coincide at some emissivity which is usually a function of temperature.

The results of the last calibration, used in most of the calculations, are shown in Figures A-11 through A-15.

FIGURE A-10

TORQUE METER CALIBRATIONS

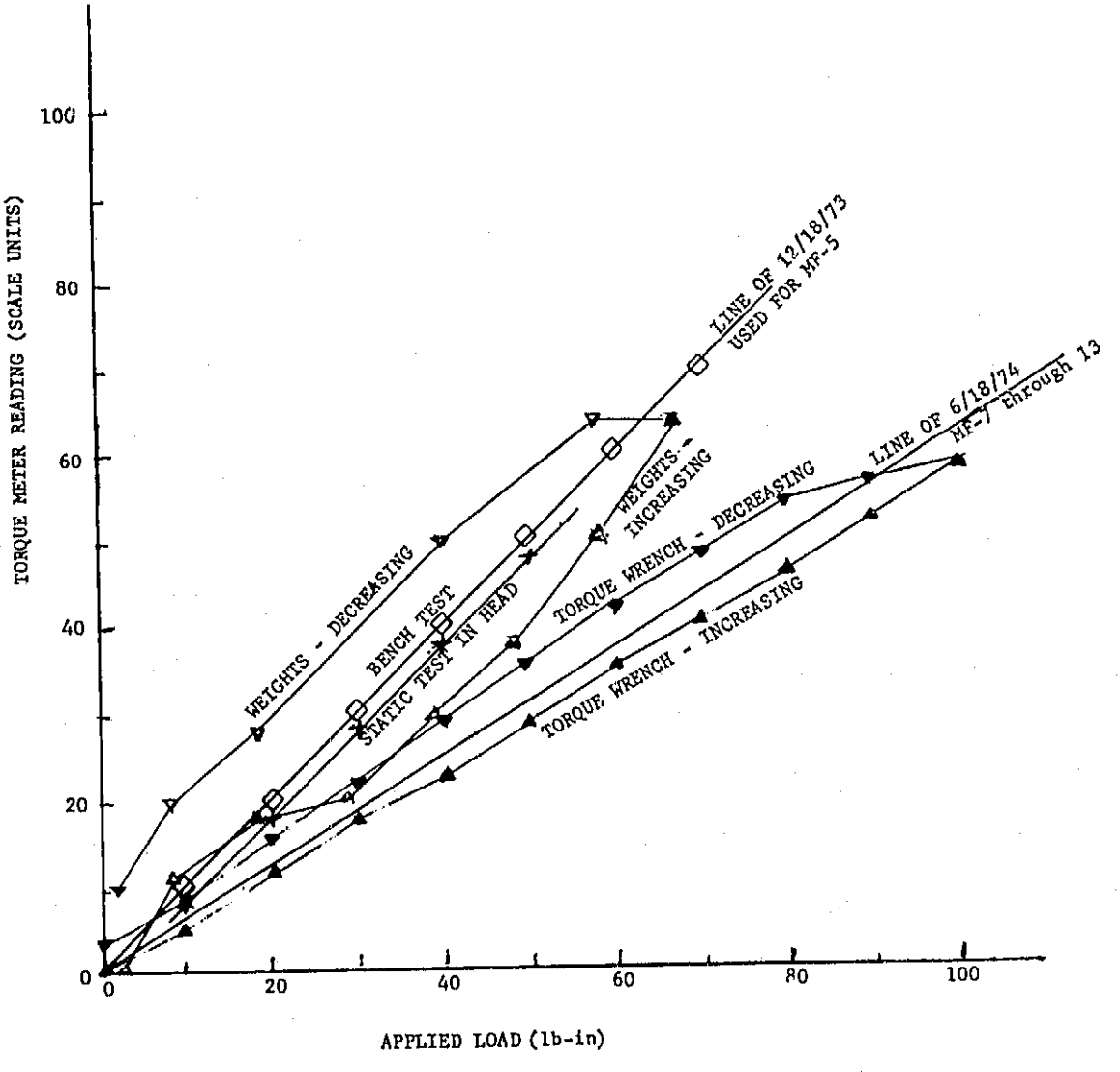


FIGURE A-11

INSTRUMENT No. 1 DATA FOR EMISSIVITY = 1.00

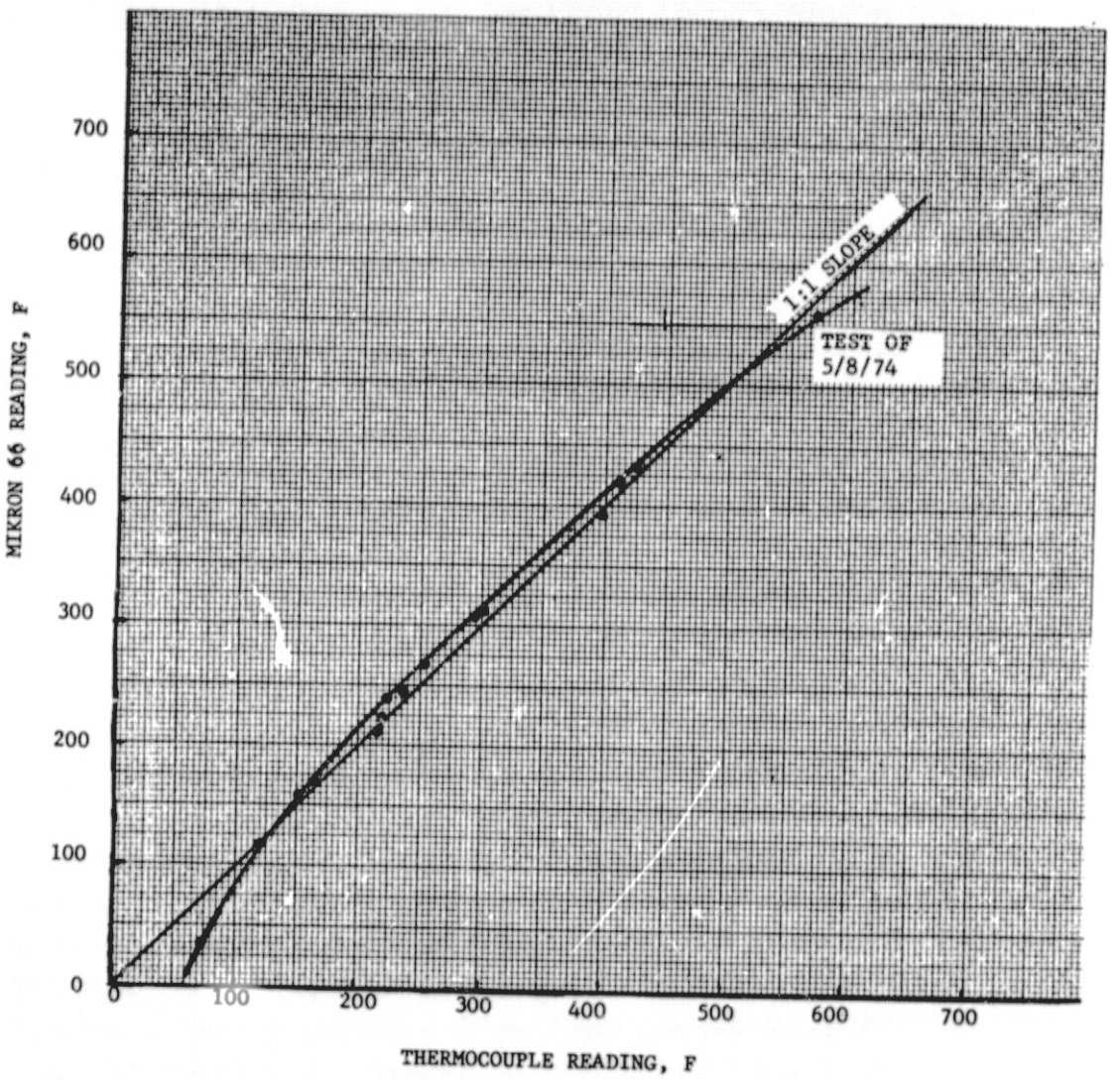


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FIGURE A-12

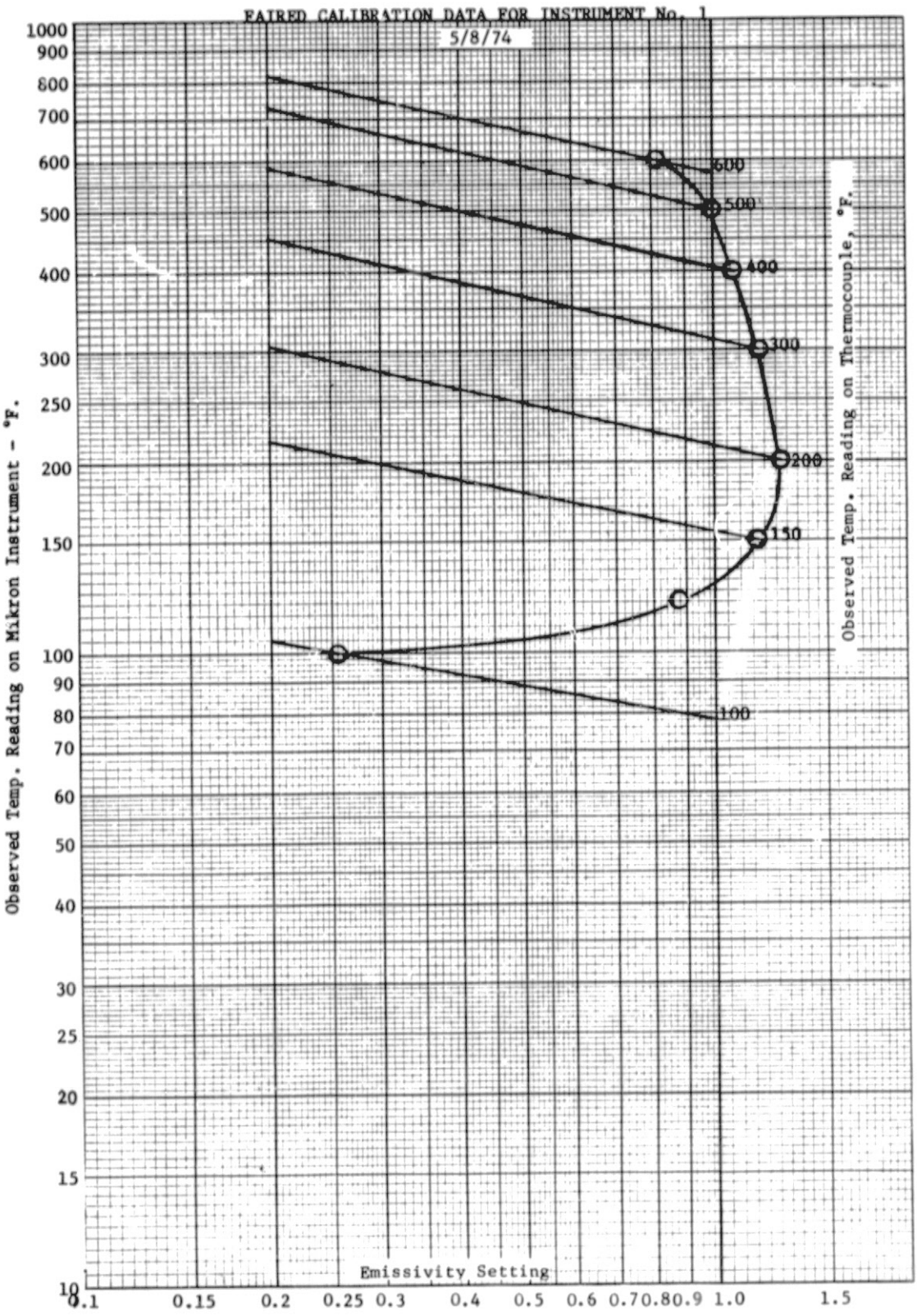
PAIRED CALIBRATION DATA FOR INSTRUMENT No. 1
5/8/74
Observed Temp. Reading on Mikron Instrument - °F.
Observed Temp. Reading on Thermocouple, °F.
Emissivity Setting
1000
900
800
700
600
500
400
300
250
200
150
100
90
80
70
60
50
40
30
25
20
15
10
600
500
400
300
200
150
100
0.1
0.15
0.2
0.25
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.5

FIGURE A-13

INSTRUMENT No. 2 DATA FOR EMISSIVITY = 1.00

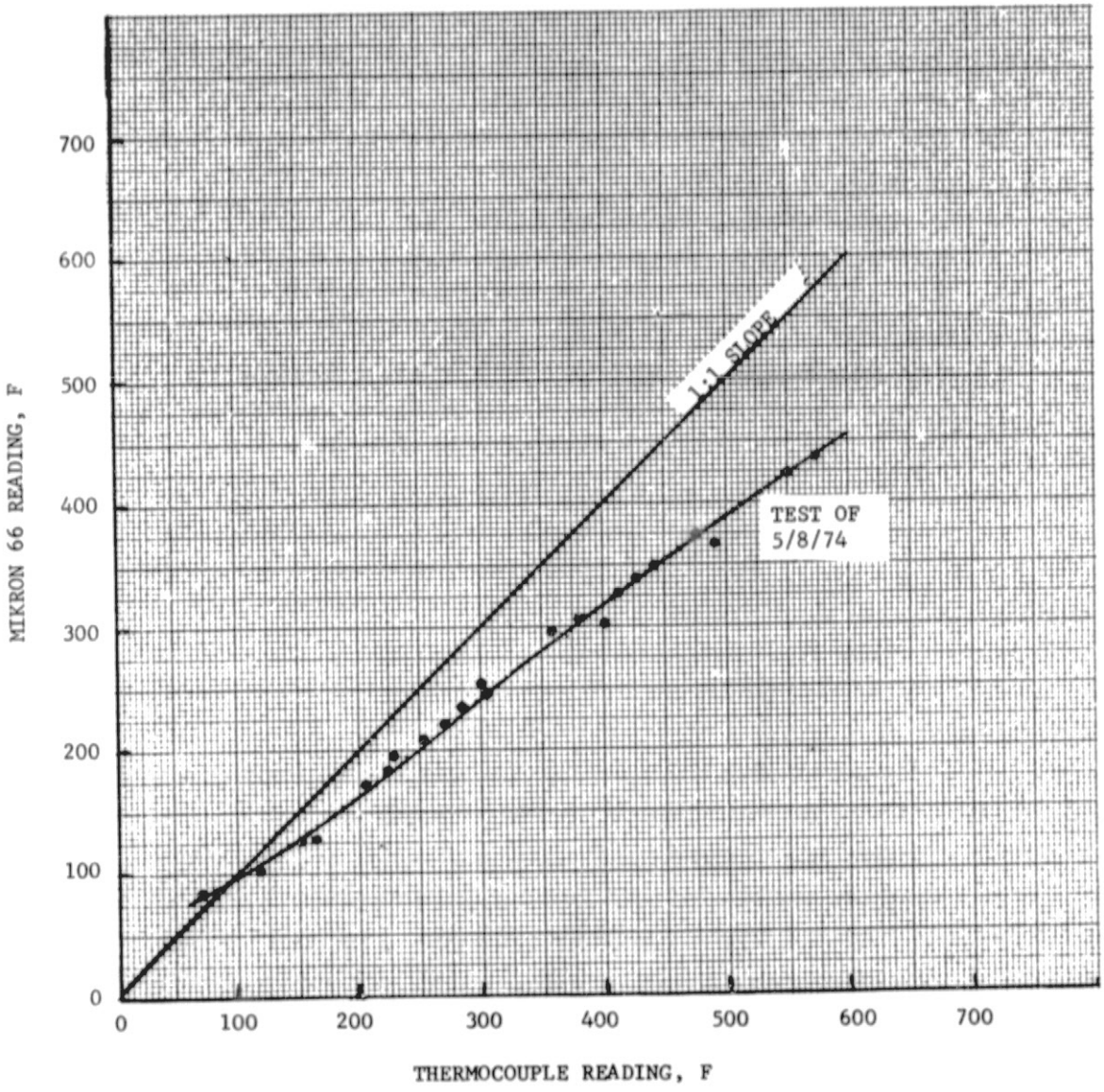

FIGURE A-14

FAIRED CALIBRATION DATA FOR INSTRUMENT No. 2

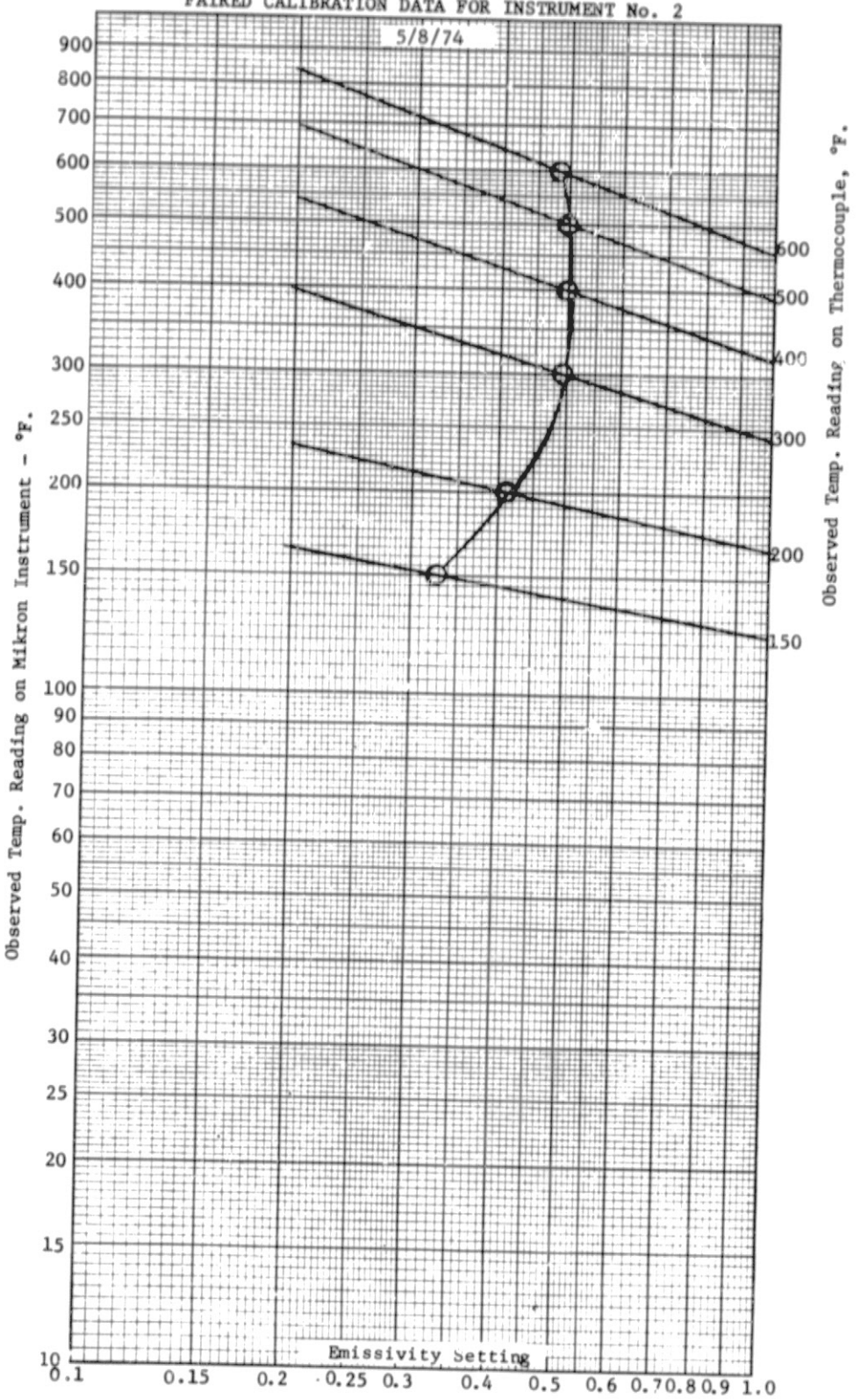

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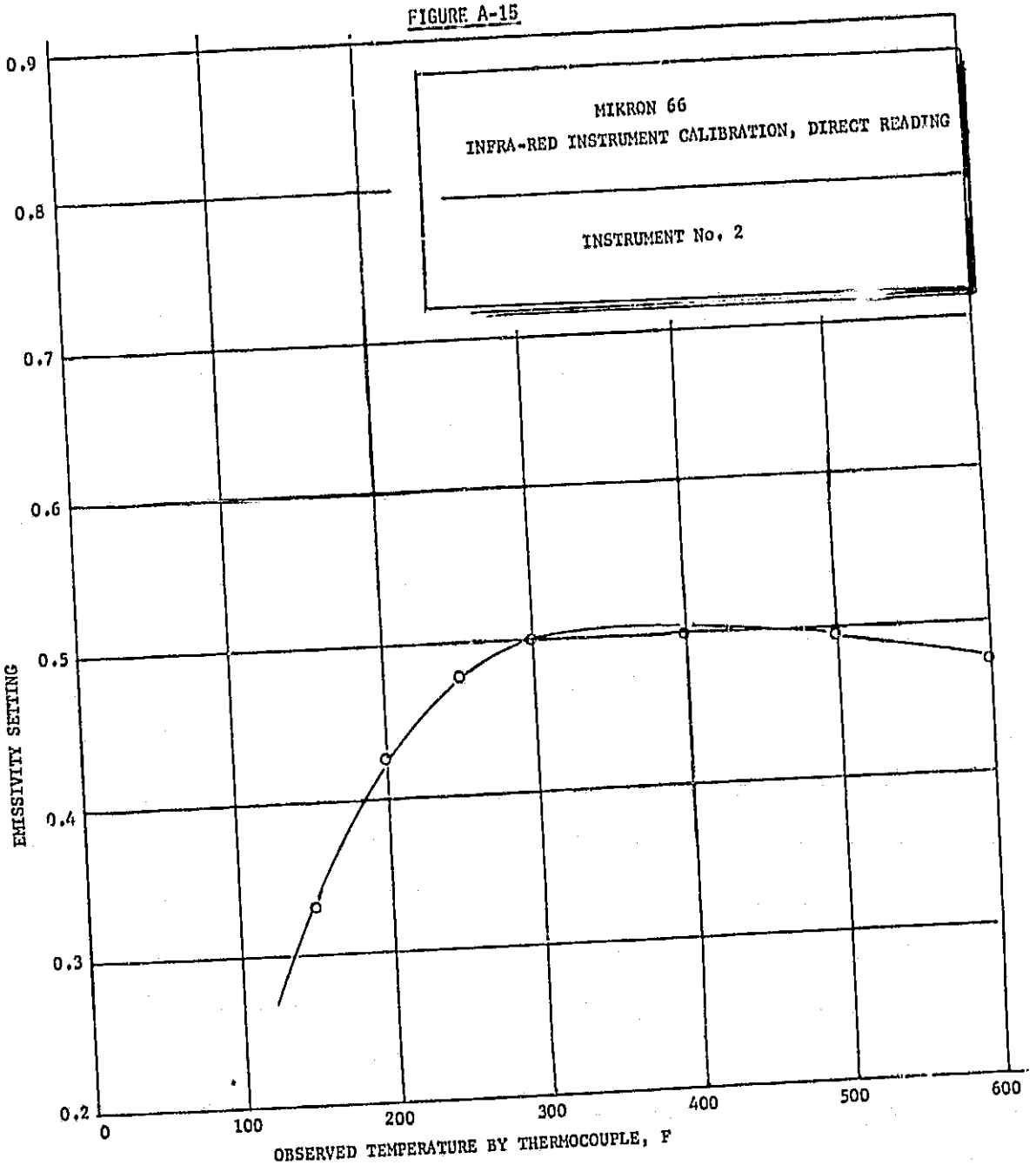
FIGURE A-15


MIKRON 66
INFRA-RED INSTRUMENT CALIBRATION, DIRECT READING
INSTRUMENT No. 2
EMISSIVITY SETTING
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0
100
200
300
400
500
600
OBSERVED TEMPERATURE BY THERMOCOUPLE, F

4. Procedure

4.1 The runs were made under the conditions shown in Tables A-1 and A-II. Some runs were not continued to thermal equilibrium, since the conclusion could be drawin without obtaining stabilized heat transfer.

4.2 Before starting rotation, the entire gearbox was preheated to 366K (200F) by flowing the cooling air and support oil at this temperature for at least 1/2 hr. The microfog generator was also brought to this temperature with heated air.

4.3 In general, gears were run-in for at least 10 minutes at minimum load. With the bypass valve (Figure A-3) open, this is about 12000 PA (1.75 psi) oil pressure which corresponds to 6900 N/m (130 lb/in) or 323 MPa (46.8 kpsi) Hertz tooth pressure.

4.4 Examination of the used gears by means of the ASTM D 1947 binocular micorscope was not very useful except in selecting the proper spacers for axial travel (see 3.2). The best results were obtained with a Talysurf (Taylor Hobson Model 3). This was used to examine those used gears on which there seemed to be some hope for obtaining useful information. However, as use of the Talysurf required disassembly of the Ryder gear head for removal of the test gears, no examinations were made between steps in a run series (Table A-11).

5. Optimization Studies

5.1 The cooling nozzle position was the subject of much pre-operational analysis, and the place chosen appears to be the only one that is logically possible under the provision. This implies both optimum cooling and the optimum point for calculating heat transfer coefficients. Obviously, the latter condition cannot be met if the air is applied at some point where the metal temperature cannot be measured, since the calculation requires metal and air temperatures at the inlet and outlet. The point for optimum cooling is, of course, that at which the tooth surface is hottest. Since the heat is all generated in the mesh, this must be at the demesh point. An additional incentive is that the demeshing gears provide pumping action, so that air must rush in to fill the vacuum left by the withdrawing teeth. Thus, all criteria agree that the air should go in at the demesh point and that tooth temperature be measured there also (see Figure A-7).

In retrospect, one experiment that might have been desirable would have been axial injection of the cooling air, analogously to the optimum microfog nozzle position determined below. Perhaps a quick check of this geometry can be included under Task II. However, a considerable amount of air will still be required in the present tangential position to purge microfog and oil slung off the gears away from the calcium floride window.

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TABLE A-1

LOG OF MICROFOG RUNS

Runs	Gears	Purpose and Conditions
MF-0	W/W (P&W)	Demonstration
MF-1	W/N (P&W)	Practice and Training
MF-2	W/N (P&W)	Practice and Training
MF-3	N/N (P&W)	Checking Spacers
MF-4	N/N (CI)	Demonstration and Data
MF-5	N/N (P&W)	Oil Starvation
MF-6	N/N (CI)	MF-4 Gears (turned over)
MF-7A	N/N (CI)	Sage Micropump, ER&E nozzle
B		Tauratron Micropump, ER&E nozzle
MF-8	W/N (CI)	Radial NASA nozzle
MF-9	W/N (CI)	Axial NASA nozzle
MF-10	W/N (CI)	Tangential NASA nozzle
MF-11 A, B, C	W/N (CI)	Step Loading Run
MF-12 A, B, C	N/N (CI)	Reduced Cooling Air
MF-13 A, B, C	N/N (CI)	High Speed Runs

W = wide

N = narrow

P&W = Pratt and Whitney Division, United Aircraft

CI = Clipper Industries

TABLE A-II

SUMMARY OF MICROFOG RUN CONDITIONS

Run No.	Speed krpm	Load MPa	Load Kpsi	Oil Rate cm^3/hr.	Oil Rate oz./hr	Cooling Air[1] mols/s	Cooling Air[1] SCFM	Nozzle Position	Duration Min.	Wear Rate um/s	Wear Rate mils/hr	Heat Transfer Stabilized
MF-7A	10	690	100	19.2	0.65	0.845	26	Radial	35	Not Read		Yes
MF-8	10	1380	200	13.8	0.47	0.650	21	Radial	10	0.021	2.9	No
MF-9	10	1380	200	24.0	0.81	0.610	29	Axial	10	0.016	2.3	No
MF-10	10	1380	200	28.2	0.95	0.630	30	Tangential	10	0.025	3.5	No
MF-11A	10	1035	150	8.7	0.29	0.630	30	Axial	10	Not Read		No
MF-11B	10	1035	150	5.4	0.18	0.610	29	Axial	10	Not Read		?
MF-11C	10	1380	200	5.4	0.18	0.610	29	Axial	18	0.042[2]	5.9[2]	Yes
MF-12A	10	1380	200	7.8	0.26	0.610	29	Axial	10	Not Read		No
MF-12B	10	1380	200	7.8	0.26	0.360	17	Axial	17	Not Read		Yes
MF-12C	10	1380	200	7.8	0.26	0.250	12	Axial	8	Burnt		No
MF-13A	10	1380	200	7.5	0.25	0.570	27	Axial	17	Not Read		Yes
MF-13B	12.5	1380	200	7.5	0.25	0.570	27	Axial	7	Not Read		Yes
MF-13C	12.5	1380	200	7.5	0.25	0.380	18	Axial	11	Not Read		Yes
MF-13D	12.5	1380	200	7.5	0.25	0.315	15	Axial	16	0.024[2]	3.3[2]	?

(1) Does not include mist and boost air

(2) Assuming all wear in last period

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5.2 The microfog reclassifier nozzle position was optimized in a brief series of runs, MF-8, 9 and 10. Run MF-7 had established the ability to run at 1380 MPa (200 kpsi) so this severity was selected to maximize any effects of geometry changes. The details are shown in Table A-III.

5.2.1 The radial position was used first as it has proved satisfactory in the preliminary run. This placement was on the driven gear π rad (180°) from the mesh point. Run MF-8 used a little less oil than had been planned, but otherwise was on target.

5.2.2 The axial position was selected on the basis that it most closely simulated the work done under previous NASA contracts (1, 2, 3, 4, 5). Despite the fact that it is completely unconventional in microfog lubrication of gears, there was some theoretical justification since this position surely has the maximum capability for sweeping the spaces between teeth free from stagnant hot air. The nozzle was canted 0.25 rad. (15°) in the direction of gear rotation to direct the reclassified oil drops onto the metal, and was placed 1.57 rad (90°) before mesh as a matter of convenience to avoid making other holes in the gear case cover.

5.2.3 The tangential position was the most conventional one, being 1.05 rad (60°) before mesh with flow in the direction of gear rotation as recommended by Norgren.

5.2.4 Three criteria were applied to the optimization. These were shaft torque, gear wear rate and amount of heat gained by the air passing through the gearbox (H_2). All three should correlate with the coefficient of friction and thus be measures of the efficiency of lubrication. As shown in Table A-III, these did not agree perfectly. The axial position showed the lowest torque and wear rate, but caused the medium rate of heat pick-up by air. The less reliable infra-red temperatures, T_b and T_s, tended to confirm the indication from air heating (H_2) that the axial position was not optimum. However, it was decided that torque and wear rate were the most reliable and important criteria.

Second best position was awarded to the radial placement, on the basis of torque and wear rate. Though T_b and H_2 had shown radial as best, T_s showed it as worst. All criteria except T_s agreed that the conventional tangential placement was worst.

An interesting observation was that the tangential position produced wave marks on the rear wall of the box. Calculation showed the 3 marks correspond to 3 pulses of mist "stored" in the tooth gap.

5.3 There was little guidance available as to lubricant requirements. The Norgren engineers advised that a rate of 0.6 cm^3/hr would be recommended for these gears at speeds up to 2000 rpm, but they had no experience above that. Preproposal work had used 190 cm^3/hr, obviously far more than was needed. Preliminary runs had shown feasibility of running at 1380 MPa (200 kpsi) with about 20 cm^3/hr.

TABLE A-III

OPTIMIZATION OF MICROFOG NOZZLE PLACEMENT

Run No.	Nozzle Position	Oil Rate cm^3/hr	Oil Rate oz./hr	Torque N m	Torque lb-in	Wear Rate u m/s	Wear Rate mil/hr	Air Heat Watts	Air Heat BTU/hr	T_b K	T_b F	T_s K	T_s F
MF-8	Radial	13.8	0.47	11.49	101.7	0.021	2.9	566	1930	471	388	428	310
MF-9	Axial	24.0	0.81	10.93	96.7	0.016	2.3	627	2140	495	432	388	238
MF-10	Tangential	28.2	0.95	11.87	105.0	0.025	3.5	650	2220	528	490	369	205

GENERAL CONDITIONS

Load = 1380 MPa (200 kpsi) Hertz

Speed = 10 krpm

Air Inlet = 362 $\pm$ 3K (192 $\pm$ 5F)

Mist Air = 0.023 mols/s (1.1 SCFM)

Boost Air = 0.026 mols/s (1.25 SCFM)

The optimization experiments are shown in Table A-IV, though MF-12A was not planned to be part of the series. In addition, the fact must be recorded that a zero oil rate was tested on Run MF-5A when the oil pump stalled. The gears were scuffed in a few seconds time, but could be reversed and were reused in MF-5B. The results at 1380 MPa (200 kpsi) in Table A-IV are not as definitive as could be wished, due to lack of a wear rate on MF-12A. However, the other criteria show that there were only minor increases in torque, T_b and a decrease in H_2 on going from 24.0 to 7.8 cm^3/hr of oil. The controversial T_s did increase, supporting the belief that the minor effects are real. The next step, from 7.8 to 5.4 cm^3/hr, caused significant increases in torque, H_2, T_B and T_S. The wear rate is significantly higher than for MF-9, and it is plausible to assume that most of this is due to the last cut in oil rate.

The above analysis provides very reassuring evidence that the variations in oil rate experienced during optimization of nozzle placement in Section 6.2 and Table A-III did not approach the critical level of about 8 cm^3/hr.

One detail not fully recognized until after completion of the test program was that the oil/mist air ratio had been varied along with the oil flow rate, as shown in Table A-IV. For comparison, the Norgren design manual (6) shows aspiration rates for several oils at various temperatures. "Very good output" corresponds to 0.60 cm^3/mol (0.025 oz/SCF) while "limited output" corresponds to 0.12 cm^3/mol (0.005 oz/SCF). Thus, MF-12A and MF-11C were run at ratios below that typical of aspiration. Even MF-13 (see 2.3 above), with 0.11cm^3/mol (0.0046 oz/SCF) was borderline. The effect of this variable on such details as condensation in the piping and non-reclassifiable microfog cannot be predicted, but Task II should be run with at least 0.3 cm^3/mol (0.013 oz/SCF). (This can also be described as "10,000 ppm").

5.4 The air consumption for a helicopter transmission running on microfog was the subject of considerable speculation before starting this program. Hence, it was considered important to perform stepwise reductions of the cooling air until T_b reached 589K (600F). This was done at 1380 MPa (200 kpsi) and two speeds, as shown in Tables A-VA and A-VB. In this series, wear was not considered to be of primary importance so only the wear after MF-13D was measured and that included all wear from MF-13A, B and C. Instead, the goal was to obtain good heat transfer coefficients. These will be discussed in the next section.

The tests at 10 krpm, MF-12A, B and C and MF-13A, show progressive increase in T_b as the cooling air flow is reduced. This is partly due to poor control of the air inlet temperature, but that had only a minor influence and (T_b - T_I) also trended upward. T_s is more erratic but follows.

5.5 At the higher speed, the trend of (T_b - T_I) is reversed, and there is no evidence to show how far the air rate must be reduced to cause distress. Clearly, the speed increase had unexpectedly beneficial effects. It was anticipated that the heat transfer would improve, and perhaps that the coefficient of friction might also be lower at high speed, but there seems to be some further effect to be explored.

TABLE A-IV

OPTIMIZATION OF OIL RATE

Run No.	Oil Rate		Oil/Air Ratio		Torque		Wear Rate		Air Heat (H_2)		T_b		T_s	
	cm^3/hr	oz/hr	cm^3/mol	oz/SCF	N·m	lb-in	m/s	mil/hr	Watts	BTU/hr	K	F	K	F
MF-9	24.0	0.81	0.290	0.0123	10.93	96.7	0.016	2.3	627	2140	495	432	388	238
MF-12A	7.8	0.26	0.075	0.0031	11.30	100.0	---	---	505	1720	502	444	447	345
MF-11C	5.4	0.18	0.056	0.0023	13.00	115.0	0.042	5.9	944	3220	659	727	554	537

GENERAL CONDITIONS

Load = 1380 MPa (200 kpsi) Hertz

SPEED = 10 krpm

Nozzle = Axial

Air Inlet = 360 ± 1K (188 ± 2F)

Mist Air = 0.026 ± 0.003 mols/s (1.25 ± 0.15 SCFM)

Boost Air = 0.030 ± 0.060 mols/s (1.40 ± 0.30 SCFM)

TABLE A-VA

OPTIMIZATION OF COOLING AIR RATE, IN SI UNITS

Run No.	Speed krpm	Total Air mols/sec	Air In T_I(K)	Air T (K)	Air (H_2) Watts	T_b (K)	$T_b - T_I$ (K)	T_s (K)
MF-12A	10	0.674	359	24	505	502	143	447
MF-13A	10	0.611	364	26	458	580	216	379
MF-12B	10	0.421	390	24	316	658	268	547
MF-12C	10	0.317	394	28	254	680+	286+	719+
MF-13B	12.5	0.611	364	18	306	528	164	369
MF-13C	12.5	0.421	378	4	21	525	147	379
MF-13D	12.5	0.358	388	-8	-72	531	143	379

GENERAL CONDITIONS

Load = 1380 MPa Hertz

Oil Rate = 7.65 $\pm$ 0.15 cm^3/hr

+ indicates subsequent rise to failure

TABLE A-VB

OPTIMIZATION OF COOLING AIR RATE, IN ENGLISH UNITS

Run No.	Speed krpm	Total Air SCFM	Air In T_I (F)	Air T (F)	Air (H_2) BTU/hr	T_b (F)	$T_b - T_I$ (F)	T_s (F)
MF-12A	10	32.0	186	44	1720	444	258	345
MF-13A	10	29.0	196	46	1560	585	389	223
MF-12B	10	20.0	242	43	1080	724	482	525
MF-12C	10	15.0	250	50	870	834+	584+	690+
MF-13B	12.5	29.0	196	32	1045	491	295	205
MF-13C	12.5	20.0	220	7	72	486	266	223
MF-13D	12.5	17.0	238	-14	246	496	258	223

GENERAL CONDITIONS

Load = 200 kpsi Hertz

Oil Rate = 0.29 $\pm$ 0.005 ox./hr

+ indicates subsequent rise to failure

6.0 Calculations

6.1 Calculation of the heat input is simply a matter of adjusting the reading to the selected units since

$$E = K_1 2\pi\theta n$$

where θ is torque and n is rpm. For SI units, $E = 0.01045\ \theta n$ watts and for English units, $E = 0.04037\ \theta n$ BTU/hr.

6.2 The air calculations begin with combining all three streams - mist, boost and cooling air. (In Runs MF-5B, and 7A the mist and boost air streams were not heated, and in MF-5B the air used to purge the IR thermometer tube was also unheated.) The total is converted from SCFM to mols/sec by the perfect gas law. This is combined with the inlet and outlet temperatures (T_I and T_2), using Figure A-16 to obtain the enthalpy change in joules/mol (H_2).

6.3 The heat carried out in the service oil (H_3) is obtained from the measured flow rate, the density of 0.830 g/cm^3 at 366 K (200F) and the specific heat of 2230 J/kgK at 366K (0.533 BTU/lb F at 200F). These oil properties were obtained from the Exxon Data Book for Designers (September 1973), Charts BA-2 and BE-2 respectively.

6.4 Heat leakage was calculated as described in Section 3.1, using

$$H_1 = 6.61\ (T_c - T_a)$$

for watts/K (12.53 $(T_c - T_a)$ for BTU/hr F).

6.5 The heat balance would normally be calculated as a percentage

$$100\ (H_1 + H_2 + H_3)/E$$

and on this basis runs better than 95% in 5 out of 9 tests. This is shown in Tables A-VIA and A-VIB as "Gross Balance". However, in this case, Gross Balance is deceptive, due to the high level of H_3. It is more meaningful to calculate "Net Balance ", and on that basis the results are far less satisfactory. "Net Balance" is defined by:

$$100\ H_2/(E - H_1 - H_3)$$

6.6 Because of the preliminary nature of this work, and the uncertainty of heat balance, no attempt was made to calculate H_W, H_B and H_S as shown in Figure A-6.

6.7 To assist in deciding the relative merits of the "Air watts" (H_2) versus "Net watts", calculations were made from a mathematical model for gear loss discussed by Shipley in Dudley's "Gear Handbook" (12).

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FIGURE A-16

TEMPERATURE–ENTROPY DIAGRAM FOR

AIR

1954

PRESSURE, INTERNATIONAL ATMOSPHERES ——
ENTHALPY, JOULES/MOLE —·—
VOLUME, CUBIC CENTIMETRES/MOLE - - - -

1 MOLE = 28·96 GRAMS

ENTROPY AND ENTHALPY ZERO FOR LIQUID BOILING AT 1 ATM AND 78·8 °K

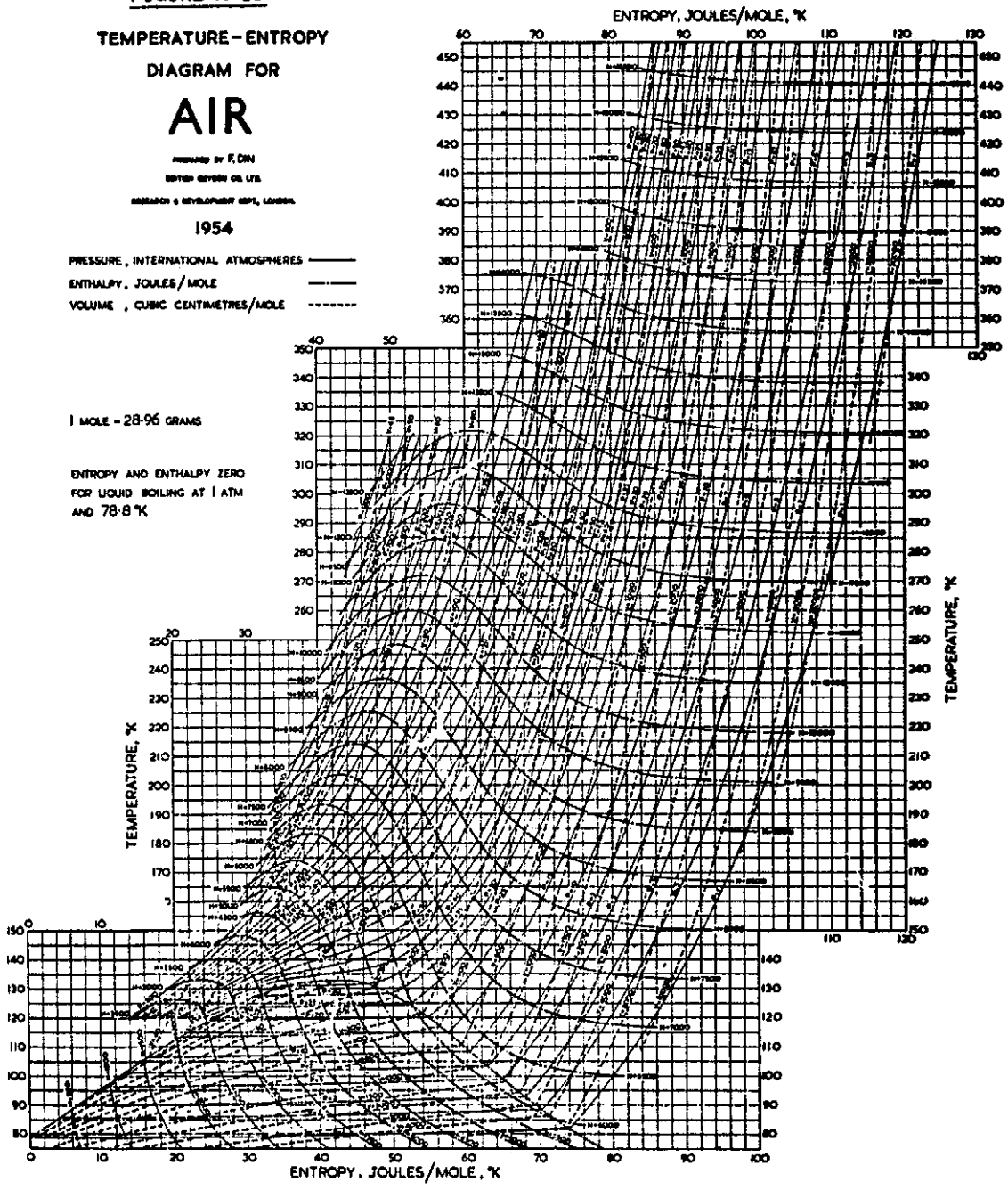

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TABLE A-VIA

HEAT BALANCE CALCULATIONS, IN SI UNITS

Run No.	Speed krpm	Load MPa	Input Watts (E)	Air Temperature T_1 (K)	T_2 (K)	Flow mols/s	Air (H_2) Watts	Oil (H_3) Watts	Leakage Watts (H_1)	Cutput Watts	Gross Balance (%)	Net Watts	Net Balance (%)	Calculated Watts (H_6)
MF-5B	10	690	7600	387	361	0.604(1)	204	6833	440	7477	98.4	327	82	160
MF-7A	10	690	9460	365	364	0.611(2)	119	8247	470	8836	93.4	743	16	160
MF-11B	10	1035	10642	361	364	0.674(3)	34	9554	462	10050	94.4	626	5	360
MF-11C	10	1380	13600	361	409	0.674(3)	944	11563	624	13131	96.6	1413	67	800
MF-12B	10	1380	13010	390	414	0.421(3)	316	10610	665	11951	89.7	1735	18	800
MF-13A	10	1380	11430	364	390	0.611(4)	458	9029	594	10081	88.2	1801	25	800
MF-13B	12.5	1380	14790	364	382	0.611(4)	306	13453	487	14340	97.0	756	40	1000
MF-13C	12.5	1380	14290	378	382	0.421(4)	21	13410	480	14011	98.0	300	7	1000
MF-13D	12.5	1380	14290	388	380	0.358(4)	-72	13089	497	13614	95.3	604	--	1000

(1) Including 0.225 mol/s of mist, boost and purge air at 302K.

(2) Including 0.0632 mol/s of mist and boost air at 300K.

(3) Including 0.0632 mol/s of mist and boost air at 366°K.

(4) Including 0.0422 mol/s of mist and boost air at 366°K.

TABLE A-VIB

HEAT BALANCE CALCULATIONS, IN ENGLISH UNITS

Run No.	Speed krpm	Load kpai	Input BTU/hr	Air Temperature T_1 (F)	Air Temperature T_2 (F)	Flow SCFM	Air (H_2) BTU/hr	Service (H_3) Oil BTU/hr	Leakage BTU/hr (H_1)	Output BTU/hr	Gross Balance (%)	Net BTU/hr	Net Balance (%)	HG BTU/hr
MF-5B	10	100	25900	226	190	28.7(1)	696	23300	1500	25500	98.4	1115	62	545
MF-7A	10	100	32300	198	197	20.0(2)	406	28100	1600	30150	93.4	2540	16	545
MF-11B	10	150	36300	190	196	32.0	116	32600	1580	34300	94.4	2140	5	1230
MF-11C	10	200	46400	190	276	32.0	3220	39500	2130	44800	96.6	4820	67	2730
MF-12B	10	200	44400	242	285	20.0	1080	36200	2270	40800	89.7	5920	18	2730
MF-13A	10	200	39000	196	242	29.0(3)	1560	30800	2030	34400	88.2	6145	25	2730
MF-13B	12.5	200	50500	196	228	29.0(3)	1040	45900	2000	48930	97.0	2580	40	3410
MF-13C	12.5	200	48800	220	227	20.0(3)	72	45800	1980	47800	98.0	1020	7	3410
MF-13D	12.5	200	48800	238	224	17.0(3)	-245	44700	2040	46500	95.3	2060	--	3410

(1) Includes 10.7 SCFM of mist, boost and purge air at 84°F.

(2) Includes 3 SCFM of mist and boost air at 77°F.

(3) Includes 2 SCFM of mist and boost air at 200°F.

This requires assuming a coefficient of friction (F) which Shipley shows to be between 0.03 and 0.05. The lower value was selected after testing both, though it probably would not be correct for the starved case MF-12C.

The equation used is

$$H_G = \frac{0.262\ WDnF}{33000} \times 746$$

and gives heat from the gears in watts when W is lb force on the tooth, D is pitch diameter (in), and n is rpm.

Comparison of H_G with H_2 and "Net watts" shows that both are probably in error, most of the time. Of course, the fact that Shipley's F = 0.03 to 0.05 was determined in conventional gearboxes offers reason to suspect that H_G may be too high. Resolution of this problem must await further experiments in a better-designed gearbox for this purpose.

6.8 In order to permit comparison with other systems, the heat balances must be reduced to heat transfer coeffieicnts. The results of these calculations are shown in Tables A-VIIA and A-VIIB, on the three bases (Air, Net and H_G) discussed above.

The first step required is estimation of the temperature of the cooling air after expansion. The pressure drop in that nozzle was about 0.2 MPa(30 psi), and some refrigeration effect might be expected. This was estimated from Figure A-16 for an adiabatic (isenthalpic) expansion. The decrease from T_I and T_1 proved to be surprisingly small - about 1 to 2K (2 to 4 F). Common experience with compressed air indicates cooling of about 5K (10F), but that is due to compression above 0.7MPa and to the greater slope of the iso-H lines at around room temperature.

The next calculation is the effective temperature difference. This is usually done by a chart such as Figure A-17. For most cases, the flow is counter-current-that is, the coolest air meets the hottest metal at the inlet. This model was assumed here. Though the mist and boost air streams did not enter at the cooling air inlet, all three were lumped together for use in Figure A-17; $\Delta T_1 = T_s - T_1$ and $\Delta T_2 = T_b - T_2$.

The effective transfer area of the gear tooth cannot be determined with any rigor. It was quite arbitrarily decided to use the entire working area of the teeth. Measurement of the worn area on a test gear showed this to be 6.0 by 6.35 mm (0.236 by 0.250 in.) per tooth, giving an area of 2.134×10^{-3} m^2 (3.31 in^2) for the 56 teeth.

C-9

TABLE A-VIIA

HEAT TRANSFER COEFFICIENTS IN SI UNITS

Run No.	Speed krpm	Load MPa	Air Data mols/sec	T_1 (K)	T_2 (K)	Gear Data T_b (K)	T_s (K)	LMTD (K)	Watts/m^2 K Air	Net	H_G
MF-5B	10	690	0.604 (1)	375	361	385	393	21	455	1730	3570
MF-7A	10	690	0.611 (2)	363	364	507	428	98	570	3555	770
MF-11B	10	1030	0.674	359	364	485	390	67	240 (3)	4380 (3)	2525 (3)
MF-11C	10	1380	0.674	359	409	659	554	222	1990 (3)	2980 (3)	1690 (3)
MF-12B	10	1380	0.421	390	414	658	547	194	765	4195	1930
MF-13A	10	1380	0.611	362	390	580	379	71	3020	12020	5280
MF-13B	12.5	1380	0.611	362	382	528	369	44	3260	8040	10660
MF-13C	12.5	1380	0.421	377	382	525	379	34	80	4130	13790
MF-13D	12.5	1380	0.358	388	380	531	379	---	----	----	----

(1) Including 0.225 mol/s of mist, boost and purge air at 302K

(2) Including 0.0632 mol/s of mist and boost air at 300 K

(3) Based on the working area only of gears.

TABLE A-VIIB

HEAT TRANSFER COEFFICIENTS, IN ENGLISH UNITS

Run No.	Speed kprm	Load kpsi	Air Data SCFM	Air Data T_1 (F)	Air Data T_2 (F)	Gear Data T_b (F)	Gear Data T_s (F)	LMTD (F)	BTU/hr ft^2 F Air	BTU/hr ft^2 F Net	BTU/hr ft^2 F H_G
MF-5B	10	100	18.0 (1)	214	190	233	248	38	80	129	630
MF-7A	10	100	29.0 (2)	194	197	453	311	177	[illegible]0	625	135
MF-11B	10	150	32.0	186	196	413	243	120	42 (3)	770 (3)	445 (3)
MF-11C	10	200	32.0	186	276	727	537	400	350 (3)	525 (3)	295 (3)
MF-12B	10	200	20.0	242	285	724	525	350	135	735	340
MF-13A	10	200	29.0	192	242	585	223	128	530	2120	930
MF-13B	12.5	200	29.0	192	228	491	205	84	575	1415	1880
MF-13C	12.5	200	20.0	219	227	486	223	61	15	1730	2430
MF-13D	12.5	200	17.0	238	224	500	235	---	---	----	----

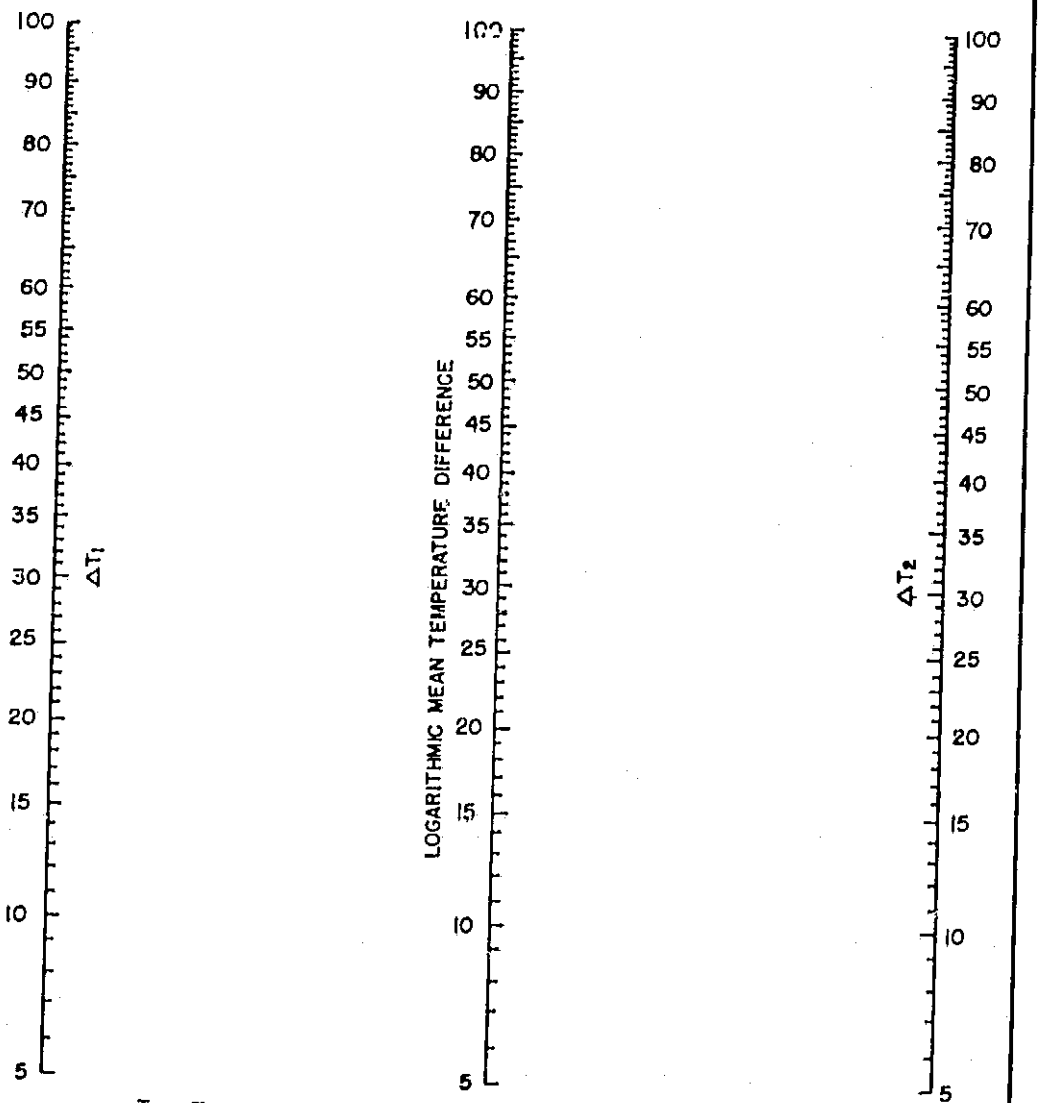
(1) Includes 10.7 SCFM of mist, boost and purge air at 84F

(2) Includes 3 SCFM of mist and boost air at 77F

(3) Based on the working area only of gears

FIGURE A-17

LOGARITHMIC MEAN TEMPERATURE DIFFERENCE



$$M.T.D. = \frac{\Delta T_1 - \Delta T_2}{\log_e \frac{\Delta T_1}{\Delta T_2}}$$

WHEN ΔT_1 AND ΔT_2 ARE NOT WITHIN THE CHART RANGE, THESE VALUES MAY BE MULTIPLIED BY A FACTOR, AS 0.5, 2, 10, OR 100, ETC.

FOR EXAMPLE:

GIVEN ΔT_1 (ACTUAL) = 200, AND ΔT_2 (ACTUAL) = 20.

USE 0.5 AS FACTOR, AND $\Delta T_1' = 100$, AND $\Delta T_2' = 10$.

FROM THE CHART, M.T.D.$'$ = 39.5

OR M.T.D. (ACTUAL) = $\frac{1}{0.5} \times 39.5 = 79.$

REFERENCE: POWER PLANT ENG. 35, 937 (1931)

Many arguments could be raised against this, or any other, as the area for heat transfer. The most confusing point is that heat is not generated uniformly along the tooth surface. As shown in Figure A-18, the generation rate is linearly proportional to the distance from the pitch circle, and it is generally accepted that the "flash temperature "(T_F) follows the same pattern. However, by the time the teeth have demeshed enough to become accessible to the cooling air and IR thermometer, this tidy pattern has been distorted by thermal conduction in the metal. The tip, pitch circle, and coast side of the gear have bled off some heat from the addendum, while the dedendum has also lost heat to the web and hub of the gear, resulting in a T_S curve shown schematically in Figure A-18. Thus, the energy which was at first confined to the little 38 mm^2 rectangle has already flowed into a larger area and will continue to do so in competition with the heat transfer to cooling air.

There might be some justification for putting the tooth tip area in, and there might also be some for leaving out the hot face of the driven gear since the cooling air strikes only the coast side.

In Runs 11B and C, only the working 25% of the wide gear was used; if the whole tooth face were used, these values would be multiplied by 0.40. Such arbitrary decisions constantly arise in heat transfer work, but in most cases they are covered by "conventions" set up many years ago by the classical workers. Such questions as were raised in the preceding paragraph must be settled by joint agreement of the present team. The important point is not what decisions are made, so much as that they be so explicit as to avoid future confusion.

6.9 Comparison with previous work is difficult because the systems are not closely similar. However, a simplified Nusselt plot is shown in Figure A-19. Although the Nusselt equation is based on the mass Reynolds number, when all other variables are held constant this becomes

$$h = C\, Q_1^m$$

where h is the heat transfer coefficient and Q_1 the air flow rate. This can be adjusted for rpm (n) by

$$h = C\, (nQ_1)^m$$

According to Report CR-120843 (4), the value of m should be 1/3, while Perry's Handbook (9) favors m = 0.58. As can be seen in Figure A-19, the present data are quite scattered, and would require m to be much larger than 1/3.

FIGURE A-18

THEORETICAL PATTERN OF TEMPERATURES ON A GEAR TOOTH FACE

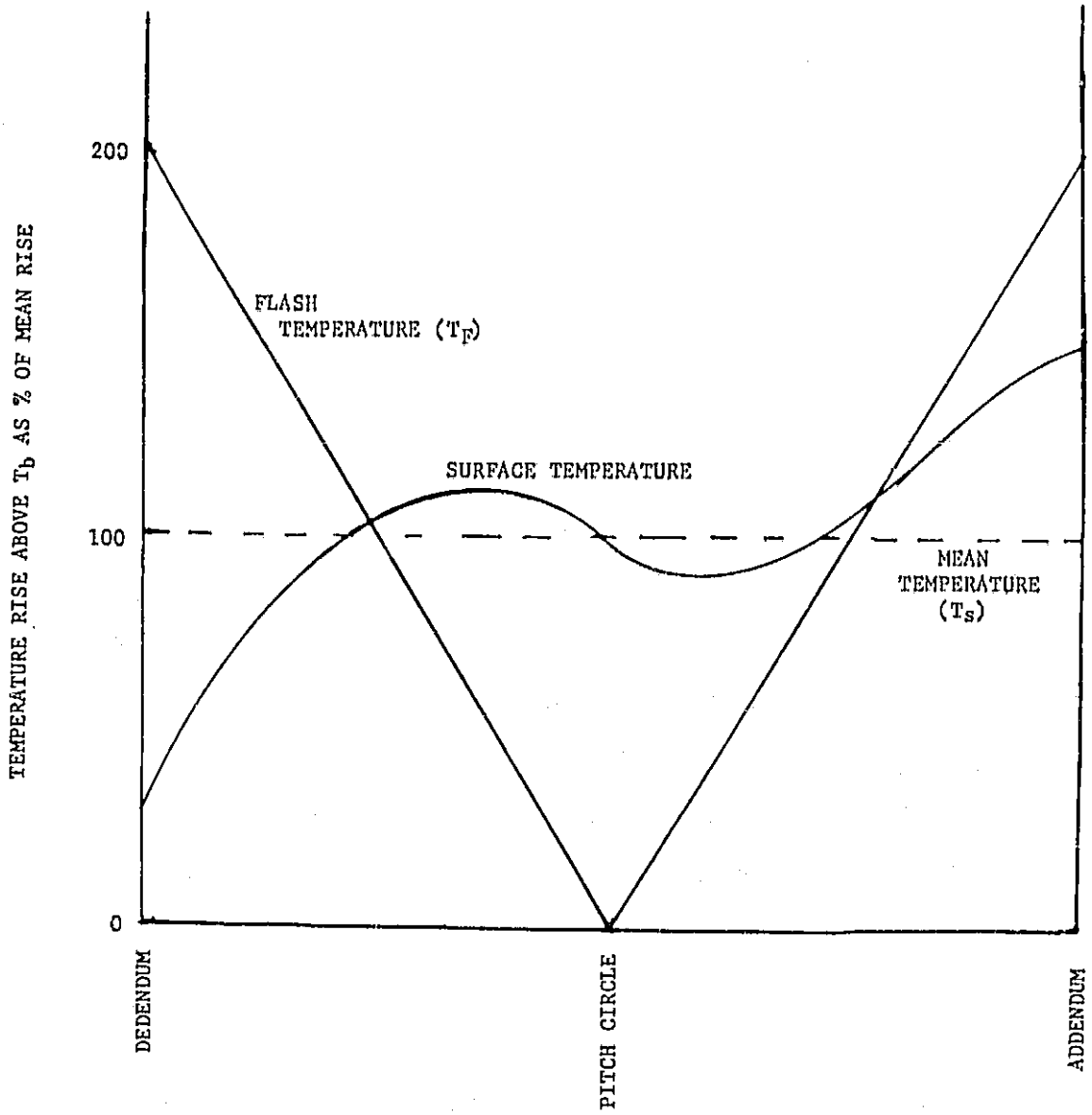


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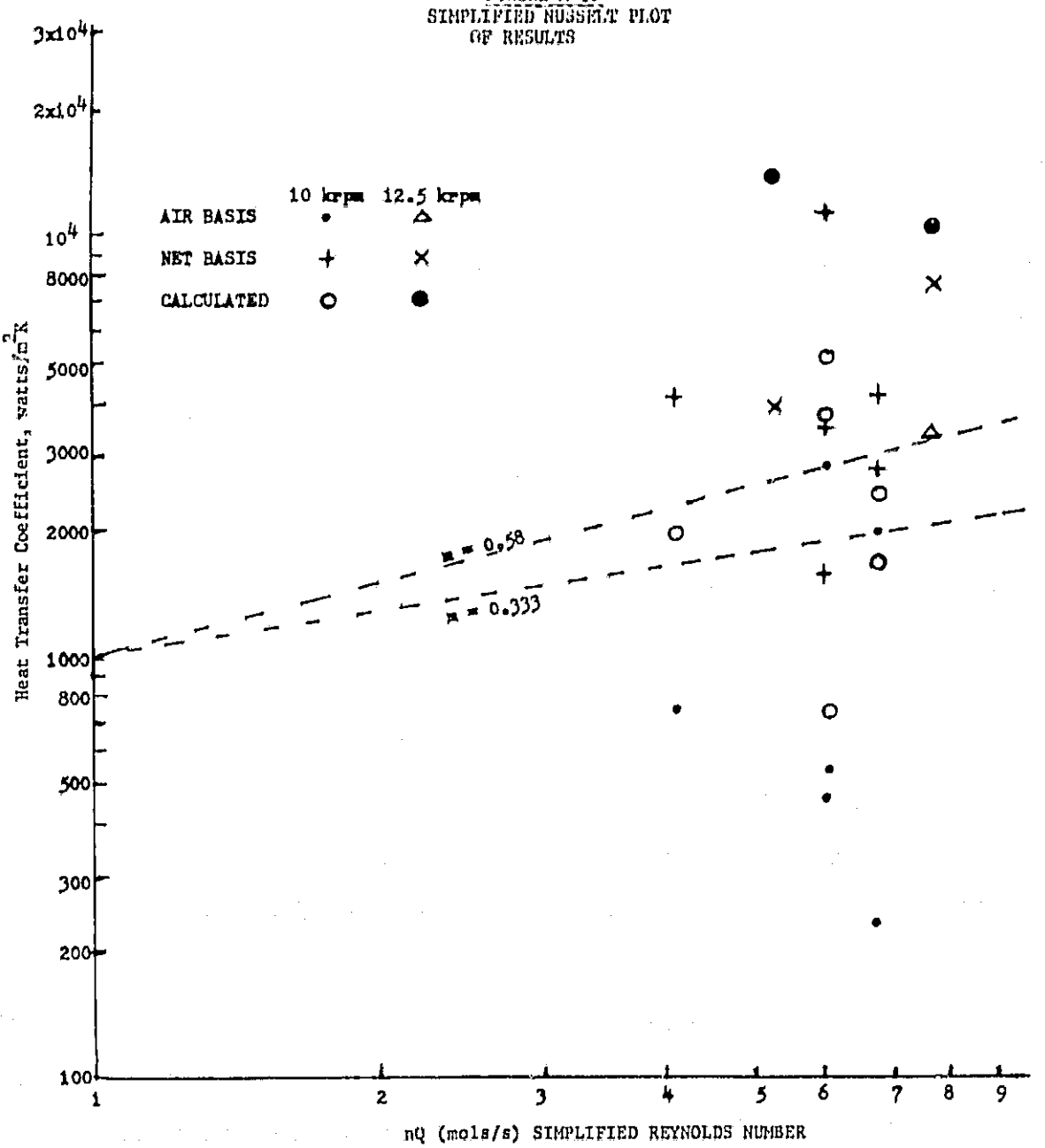
FIGURE A-19
SIMPLIFIED NUSSELT PLOT
OF RESULTS



7. Step Loading

Experiments were conducted in accordance with Task IC of the contract, but this work had to be delayed until a fair amount of optimization had been accomplished so that the data would be relevant to future Task II efforts. As a result, both time and money were exhausted before the loading could be carried to complete destruction of the gears. However, there is good reason to believe this would have occurred at the next load step of 2070 MPa (250 kpsi) Hertz. With the regular oil jet lubrication of D 1947 on P&W gears, MIL-L-23699 shows 22.5% scuffing at about 1550 MPa (225 kpsi) Hertz. Design calculations indicate the metal would show surface distress due to crushing the hardened case into the softer core at 2070 MPa, whereas through-hardened gears would tend to fail at about this load due to tooth breakage (12).

The results of the step-load tests are shown in Table A-VIII. To conserve funds, the 12.5 krpm tests were started at 1380 MPa, with such success that no lower loads were tried. For similar reasons, Run MF-7A was not repeated with the axial nozzle placement, since it would not have added significant information.

8. Gear Conditions After Tests

The contract calls for a report on gear wear and condition after tests. Talysurf charts on typical teeth from Runs MF-6B, 8, 9, 10, 11C and 13D have been forwarded to the Project Manager under separate cover. It can be stated that some roughening of all these gears took place, but no scuffing in the ASTM D 1947 sense.

All the used gears (4 wide and 12 narrow) are in storage as NASA property, awaiting shipping instructions.

9. Additional Data Requirements

Photographs of the test gears and surrounding area for Runs MF-8, 10 and 12 have been forwarded to the Project Manager. No post-test samples of system deposits nor used oil could be collected, due to the very small amount of oil used and to the vigorous scavenging effect of the cooling air. Such wear debris as was found was non-magnetic, and was probably phosphate coating from the gears.

Serious consideration should be given during Task II to using some means for recovery of debris from the exit air for examination of particle shapes and size distribution.

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TABLE A-VIII

STEP-LOAD RUNS OF TASK I C

Run No.	Speed krpm	Load MPa	Load kpsi	Gear Condition (1)
MF-7A	10	690	100	Satisfactory
MF-11A	10	1035	150	Satisfactory
MF-12A	10	1380	200	Satisfactory
MF-13B	12.5	1380	200	Satisfactory

GENERAL CONDITIONS (except MF-7A)

Oil Rate = 8.1 $\pm$ 0.5 cm^3/hr (0.27 $\pm$ 0.02 oz./hr)

Air Rate = 0.60 + 0.03 mol/s (28.5 $\pm$ 1.5 SCFM)

Nozzle Placement = Axial (2)

(1) Gear condition satisfactory indicates that visual examination justified continued testing on this surface.

(2) MF-7A was run with ER&E reclassifier nozzle located tangentailly.

IV. REFERENCES

(1) Shim J. and Leonardi, S.J., "Microfog Lubricant Application System for Advanced Turbine Engine Components", Task I Report, NASA CR-72291, (Sept. 1967).

(2) Shim, J. and Leonardi, S.J., "Microfog Lubricant Application System for Advanced Turbine Engine Components", Final Report, NASA CR-72489, (Sept. 1968).

(3) Shim, J. and Leonardi, S.J., "Microfog Lubricant Application System for Advanced Turbine Engine Components", Task I and II Report, NASA CR-72743, (May 1970).

(4) Shim, J. and Leonardi, S.J., "Microfog Lubricant Application System for Advanced Turbine Engine Components", Phase II, NASA CR-120843, (Aug. 1971).

(5) Petrucco, R.J. and Leonardi, S.J., "Microfog Lubricant Application System for Advanced Turbine Engine Components - Phase III", NASA CR-121271, (July 1973).

(6) Norgren "Micro-Fog Bearing Lubrication Design Manual NT-1", C. A. Norgren Co., Englewood, Colo. (May 1971).

(7) Howell, P. G., "Experiments in the Aerosol Lubrication of Power Gearing", Proc. Inst. Mech. Eng., Vol. 181 PT 30, 184-191 (1966-67).

(8) O'Connor, J. J. and Boyd, J., "Standard Handbook of Lubrication Engineering", McGraw-Hill, New York, 1968.

(9) Perry, R. H., "Chemical Engineers' Handbook", 4th Ed., McGraw-Hill, 1963.

(10) Bodensieck, E. J., "Specific Film Thickness - An Index of Gear Tooth Deterioration", Preprint, Aerospace Gear Comm., AGMA Meeting, Sept. 13-14, 1965.

(11) Akin, L. S., "An Interdisciplinary Lubrication Theory for Gears", Jour. Eng. for Industry, Trans. ASME, B95, No. 4 (1973).

(12) Dudley, D. W., "Gear Handbook", McGraw-Hill, New York, 1962.

(13) Brehmer, J. R., "Oil Mist Lubrication on Drive Gearing", Schmiertechnik Tribologie, ST4/71, 137-139 (Aug. 1971).

(14) Freestone, R. M., "Mist Lubrication", Lubrication (Texaco, Inc.) 54, 10, New York (1968).

(15) Henrikson, K. G., "Mist Lubrication", Lubr. Eng. 25, 69-74 (1969).

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OF POOR QUALITY

(16) Hartmann, L. M., "Lubricating Oil Requirements for Oil Mist Systems", Lubr. Eng. 28, 21-25 (1972).

(17) Lyth, W., "The Vortex Generator, Newest Approach to Oil Mist", Lubr. Eng. 27, 259-266 (1971).

(18) Ariura, Y., Ueno, T., Sunaga, T. and Sunamato, S., "The Lubricant Churning Loss in Spur Gear Systems", Bull. JSME, 16, No. 95, 881-891 (1973).

(19) Townsend, D. P. and Zaretsky, E. V., "A Life Study of AISI M-50 and Super Nitraloy Spur Gears With and Without Tip Relief", ASME Paper No. 73 - Lub - 38.

(20) Akin, L. S., Mross, J. J. and Townsend, D. P., "Study of Lubricant Flow Phenomena in Spur Gears", ASME Preprint 77-Lub-3 (October 1974).

V. SYMBOLS

D = pitch diameter
E = shaft input energy
F = coefficient of friction
h = heat transfer coefficient
H_1 = heat leakage from case to room
H_2 = enthalpy change of combined air streams in gearbox
H_3 = heat carried out in service oil
H_B = heat generated in bearings
H_G = calculated heat from test gear friction (page 35)
H_S = heat from slave gear friction
H_T = heat from test gear friction
H_{WS} = heat from slave gear windage
H_{WT} = heat from test gear windage
m = exponent in Nusselt equation
n = revolutions per minute (rpm)
P_I = pressure of air at inlet to cooling nozzle
Q_1 = flow rate of microfog (mist + boost) air
Q_2 = flow rate of cooling air
Q_3 = flow rate of service oil
T_1 = inlet temperature of microfog
T_2 = outlet temperature of combined air streams
T_3 = inlet temperature of service oil
T_4 = outlet temperature of service oil
T_a = ambient temperature in room
T_b = bulk temperature of gear
T_c = case temperature
T_F = flash temperature at tooth surface (theoretical)
T_I = air temperature at inlet to cooling nozzle
T_s = surface temperature of gear (averaged by IR measurement)
W = force of the tooth
θ = torque on input shaft